State of California

THE RESOURCES AGENCY

Department of Water Resources

BULLETIN No. 164

TEHACHAPI CROSSING DESIGN STUDIES

Book I

MAY 1965

HUGO FISHER
Administrator
The Resources Agency

EDMUND G. BROWN
Governor
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WILLIAM E. WARNE
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FOREWORD

For many years the engineers of California have studied the various routes to bring water from the South San Joaquin Valley into the southernmost part of the state. Included were alignments directly crossing over or through the Tehachapi mountain range as well as along the coastal counties of Santa Barbara and Ventura, where the apparently insurmountable height of this land barrier, as far as building an aqueduct is concerned, might be avoided. However, from the overall engineering and economic standpoint, a direct crossing of the Tehachapi range was considered feasible as well as desirable. One might say that this is another challenge to man's courage and modern-day technology not to be evaded.

After four painstaking years of relentless study, the engineers have reached their moment of decision as to the final alignment of the aqueduct as it traverses over the Tehachapi Mountains, and the mechanical scheme by which the continuous flow of 4100 second-feet of water must be lifted to a height of nearly 2000 feet.

Many alternatives were considered and supported with conviction by the Department's own staff engineers and the engineers of the Metropolitan Water District in Southern California, who will be the prime recipient of the water delivered by the system. In recognition of seismic conditions which have a direct influence upon the design of the aqueduct crossing, the Department engaged a special board of advisers on earthquake analysis made up of some of the country's most knowledgeable men in the field. To assist the Department in making detailed studies on pumps, the firm of Daniel, Mann, Johnson, and Mendenhall of Los Angeles was also engaged on a consultant basis. The

Bechtel Corporation of San Francisco was engaged by the Metropolitan Water District to assist in their overall studies.

In order to assure against any unwarranted influences on the part of its own design staff and with determination to achieve an unbiased analysis of all technical considerations brought to bear, the Department created an advisory board known as the Tehachapi Crossing Consulting Board. Its assignment was to weigh the pros and cons offered by the various factions. For six days, beginning on May 3, 1965, this board held its final hearing from the proponents of the several schemes which have survived crucial analysis up till the eve of this decision. On May 8, 1965, this board made its final recommendation to the Chief Engineer of the Department of Water Resources.

Bound between the covers of this bulletin, which has been separated into four books because of its bulk, is a compilation of the reports and resumes used to support statements and proposals made at this final board hearing. With the permission of parent authors, the original page numbers and corresponding Tables of Contents of the individual reports have been integrated in this publication for the sake of continuity. Certain proprietary information furnished by manufacturers has been deleted in respect to the conditions under which it was furnished, viz., that it not be disseminated outside the Department. Otherwise, the basic material, intentionally, has not been edited in any manner and is presented in the chronological order by which the speakers made their presentations.

We offer this bulletin as a dedication to all those who have worked tirelessly and diligently to achieve an optimum decision on this

formidable and yet exhilarating feat of engineering. We especially acknowledge the services of the Tehachapi Crossing Consulting Board whose members are listed as follows:

John R. Hardin, Chairman
Russell G. Hornberger
Thomas M. Leps
E. C. Marliave
John Parmakian
Louis G. Puls
Robert Sailer

Sacramento, California
June 1965

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BECHTEL STUDIES OF TEHACHAPI PUMP LIFT

presented to
CALIFORNIA DEPARTMENT OF WATER RESOURCES
and the
TEHACHAPI CROSSING CONSULTING BOARD
by

M.L. Dickinson, Chief Hydraulic Engineer,
Bechtel Corp. - May 5, 1965

INTRODUCTORY REMARKS

In October 1964 Bechtel and its consultants appeared before this Board to present the results of our studies of the Tehachapi Pump Lift up to that time. Our program encompassed two principal subjects, namely:

Phase 1 - A Study of the engineering feasibility and cost of alternative routes and arrangements of the pump lift.

Phase 2 - A Study of the performance to be expected from various types of pumps and associated equipment that might be suitable for alternative one-, two- and three-lift arrangements. Phase 2 was divided into two parts:

(a) Research into the performance of existing pump installations, and

(b) Tests of model pumps in an independent hydraulic laboratory.

At the time of reporting to you last October, neither of these phases was complete. Indeed, the model test work under Phase 2b was just getting under way. The results of our work were presented in a first Interim Report dated September 1964, copies of which you have.

As the background for the presentation today, the following brief extracts from my remarks before this Board on October 22, 1964 expressed our most important conclusions as of that date:

"6. Based upon the Bechtel studies and research to date, and in consideration of the extreme importance of the Tehachapi system to the continuing growth and economic development of Southern California, it is believed that high dependability is the most important factor in selecting the most suitable location, arrangement and design of the Tehachapi Pump Lift, and that, within reasonable limits, the consideration of dependability should out-weigh both construction cost and annual operating cost.

"7. Based upon all of the forestated conclusions, it is believed that a two-lift system, either along Pastoria Creek, or along the ridge, would best meet the most important objectives of: (1) safest and most dependable design, (2) lowest first cost, and (3) lowest operating cost.

"8. It is believed that any single-lift arrangement would be less dependable in operation, would be as hazardous as regards safety against catastrophic occurrances, would be at least as expensive in both initial cost and operating cost, and would have more and greater factors beyond the limits of current knowledge and experience, than would a two-lift system."

As we proceed, it will be apparent that our tentative conclusions of last October have not been changed in any major respect. Rather, they have been verified and strengthened by substantial additional evidence. The conclusions and recommendations presented today are no longer tentative - they are specific and positive with our full confidence in their validity.

Following the October 1964 appearance before you, our studies have continued and have reached or are nearing completion in some phases. In January 1965, additional results were presented in a series of Second Interim Reports and letters, copies of which you have. For sake of brevity, in our discussions today, we will not repeat the information presented in these reports but will refer to the following on the assumption you have them available for reference:

"Second Report on European Pump Practice", by Professor Hans Gerber and Dr. Robert A. Sutherland.

"Report on Survey of American Practice in Large Capacity High Head Centrifugal Pumps and Pump Turbines", by Ray S. Quick.

"Second Interim Report on Alternative Schemes for the Tehachapi Crossing", by Bechtel Corporation.

Letter report to R.A. Skinner, dated February 23, 1965, on subject, "Tehachapi Pump-Lift System".

In addition to the above, supplementary reports and written statements incorporating the most recent information and results available as of now will be presented by Bechtel Consultants and Staff, as follows:

"European Pumping Practices"
Prof. Hans Gerber, Consultant to Bechtel

"American Pumping Practices"
Ray S. Quick, Consultant to Bechtel

"Model Test Program at NEL"
Prof. Lawrence C. Neale, Consultant to Bechtel

"Site Geology"
Cole R. McClure, Senior Supervising Geologist, Bechtel

"Alternative Pump Lift Schemes"
H.C. Hart, Project Engineer, Bechtel

A brief period has been allowed for discussion following each statement with the intent that further discussion will be held after completion of the prepared statements. We hope to cover this ground before lunch, so as to leave the afternoon open for overall evaluation and discussions.

Accordingly, to start our presentation, I will ask Professor Hans Gerber, who appeared before you last October, to summarize his views.

SUMMARY, EVALUATION, CONCLUSIONS AND RECOMMENDATIONS

SUMMARY

The remarks of Professor Gerber, Mr. Quick and Professor Neale have summarized for you the results of our research studies to date concerning the performance to be expected from the various types of pumps. You will note that our research into the performance of existing pump installations has been completed, except as it may be extended in the future as major new installations begin operation. Likewise, our program of testing at NEL certain selected existing pump models is now essentially complete. The continuing program for testing under full head large-size models specifically designed for a two-lift system at Tehachapi is just getting under way.

The other major phase of our program, the study of engineering feasibility and cost of alternative schemes, has been summarized by Messrs. McClure and Hart. Recently these studies have been carried forward to incorporate the results of recent field and office investigations undertaken by the State DWR. In this effort, we have examined the DWR test pits, trenches and core borings and made our own evaluation of this evidence. In addition, we have supplemented the DWR investigations by certain additional explorations at sites, where we considered this desirable. Since this work is not fully complete, the results presented here today cannot be considered completely final. However, it is our opinion that the total evidence available is sufficient to support the conclusions presented.

For the purpose of discussing the results of our pump research and pump lift system studies, I propose to divide the subject into its two principal parts, namely

1. Alternative types of pumps, and

2. Alternative schemes for the pump lift.

EVALUATION OF ALTERNATIVE TYPES OF PUMPS

To aid in the discussion of alternative types of pumps we have prepared enlarged diagrams which are somewhat simplified versions of Figures 1 to 5, presented in Chapter V of our Second Interim Report, dated January 1965.

Referring first to the diagram of the Lunersee five-stage pump, although a similar pump for a single-lift system at Tehachapi would have one less stage, the extreme complexity of this type of pump is apparent. Consider overall length or height of this pump. It obtains its stiffness from the complex casing (shown in grey), which is free-standing, being supported vertically at the bottom and laterally by anchors to concrete brackets at the sides. The long shaft (shown in yellow) must be of sufficient diameter to limit deflections to a very small value and to provide a conservative safety factor in critical speed. To more than double the capacity of the pump illustrated to that required for Tehachapi would necessitate increasing both the shaft and the casing sizes by large extrapolations beyond any current experience. We consider this hazardous as regards dependable results.

Note that the second and third stage impellers (shown in orange) cannot possibly be checked for accurate positioning in the casing during the original installation, nor can they ever be inspected or repaired except by dismantling the entire pump. Perhaps even more important are the seals (shown in red). Note that this pump has inter-stage seals around the impellers and the shaft. These highly vulnerable items cannot be inspected or repaired without dismantling the pump. There has been some mention that the limited head per stage of a four-stage pump at Tehachapi would minimize wear on the seals from the expected silty water. It is a fact that there would be a head of only 500 feet on the seals of stages 1, 2 and 3. However, the fourth stage outer impeller seal and outer shaft seal would be subjected to the full 2000-foot head. This is why they are larger and more complex on the section shown. Moreover, the inter-stage shaft seals and the inter-stage impeller seals are completely inaccessible for inspection or repair without dismantling the pump. In our opinion, the potential seal difficulties are one of the principal disadvantages of a four-stage pump.

Now I should like to call your attention to the two diagrams of two-stage, double-suction pumps - the Gargnano pump set in a vertical position and the Vianden pump set in a horizontal position. Except for a slight gain in economy of the pumping plant, the vertical setting has no outstanding advantage and sacrifices some advantages inherent in a horizontal setting. Therefore, let us consider only a horizontal setting, similar to the Escher Wyss pump at Vianden.

While this pump has some of the same complexities of a four-stage pump, the disadvantages are less both in number and degree. And, as Professor Gerber has pointed out, this type of pump has several substantial advantages. Its biggest advantage is probably that the hydraulic forces are precisely balanced. None of the seals must withstand full head and hence can be simple. The horizontal setting with the split casing permits accurate checking of the positioning of impellers and seals during initial installation and greatly facilitates access for inspection and maintenance during operation. Furthermore, this pump is substantially more efficient than a four-stage, single-flow pump. And finally, it has been proven by long experience with many satisfactory installations within the range of both head and capacity required for a Tehachapi two-lift system.

Finally, let us take a look at a single-stage pump, such as that at Grand Coulee. Note the extreme simplicity. The short shaft does not extend into the waterway and hence can be made as stiff as desired without affecting the impeller and casing design. There are no inter-stage seals or bearings. The single impeller can be inspected readily through a manhole and can be removed easily and quickly for any maintenance work required. The entire suction tube and volute is encased in concrete, adding a great mass for overall rigidity and protection against vibration. And, finally, this type of pump is both the least expensive and the most efficient of all.

Two major objections have been voiced against the use of a single-stage pump for a two-lift system at Tehachapi; namely: (1) it is beyond the experience of operating head in existing installations, and (2) the impeller and shaft outer seals must withstand full head.

It is a fact that no existing large single-stage pumps operate under heads of 1000 feet. However, as Mr. Quick has pointed out, one much larger and more complex reversible pump turbine operates successfully under a head of nearly 900 feet and several other large pump turbine units are in process of manufacture or installation up to heads exceeding 1200 feet. These involve both the principal American and the principal European manufacturers. In our recent visit to the Cruachan project in Scotland, Mr. Hinds, Mr. Quick and I inspected the installation of the 100 MW reversible single-stage pump turbines for a head of over 1200 feet. These were produced by a Sulzer - English Electric combination.

Let us consider also the great similarity between a single-stage pump and a Francis Turbine. Many of the mechanical and structural problems are identical. They have been solved and proved by successful operating experience of many high-head Francis turbines operating under heads equal to and higher than that for a two-lift Tehachapi installation. For example, this diagram shows the Fionnay turbine, which operates under a head of nearly 1500 feet. Because of the high head, this is a more sophisticated design than ordinary turbines. Note that the impeller and shaft seals are larger and more complex than those, for example, of the Grand Coulee pump. It should be pointed out that the seals are designed to suit the head, with a greater length so that the flow velocity through the seal (and the consequent rate of wear) is kept to a safe minimum. But even for this unusually high head, the single-stage type unit is still by far the simplest.

In our opinion, it is significant that all of the principal American and European large hydraulic equipment manufacturers either are, or are endeavoring to become, involved in producing high-head, single-stage pump turbines. Furthermore, they have indicated their interest and confidence in designing and manufacturing single-stage pumps for a Tehachapi two-lift system. There is a definite trend toward the single-stage type of unit to gain the many advantages of simplicity, economy and high efficiency. We do not believe that all of these experts can be completely wrong.

Based upon the evidence of our investigations, Bechtel is convinced that a highly satisfactory single-stage pump for a two-lift system can be produced successfully and with complete confidence in its selection at this time.

CONCLUSIONS AND RECOMMENDATIONS RE PUMPS

On the basis of the previous reports referred to above and the supplemental reports and discussions presented here today, Bechtel's conclusions and recommendations concerning alternative types of pumps for the Tehachapi pump-lift are as follows:

1. Our first and foremost recommendation is similar to that in our letter of February 23, 1965 to Mr. Skinner, which stated:

"It would not be prudent to adopt a single-lift system utilizing four-stage pumps for a system of such great importance as the Tehachapi pump-lift".

We now recommend without reservation that a single-lift system utilizing four-stage pumps should not be adopted under any circumstance. Because of the importance of this recommendation, I should like to summarize the principal reasons on which it is based:

a) Development of a four-stage pump of the characteristics planned would require a large extrapolation in specific speed beyond the limits of existing satisfactory multi-stage pumps. This would involve substantial risk that the objectives of high efficiency and dependability might not be attained.

b) An additional extrapolation in capacity to more than double that of existing multi-stage pumps would be required, entailing a large increase in the size of the shaft and in the size of the casing. The adequacy of this great extrapolation beyond existing designs could not be substantiated by model tests and is too hazardous to be prudent for this important installation.

c) The expected silty water at Tehachapi imposes the possibility of severe erosion and wear problems on the seals. The inter-stage seals cannot be inspected or repaired without great difficulty. The fourth stage outer seals must withstand the entire 2000 ft. head, double that of any alternative pump under consideration.

d) Since the inter-stage impellers and seals are inaccessible, accurate positioning cannot be checked during initial installation nor can the increased clearances due to wear be checked during operation without dismantling the entire pump.

e) Dismantling this pump for periodic inspection or maintenance is much more difficult and time consuming than for any other type of pump.

f) A four-stage pump inherently is the most complicated of any of the pump types under consideration.

g) A single-lift system with pressures of nearly 2000 ft. involves the most risk in indeterminate factors in design and construction of the high-head components of the system.

h) Other simpler, more efficient, more dependable and less expensive types of pumps are available for alternative pump lift schemes.

2. For a two-lift system, it is our opinion that either a two-stage, double-flow, horizontal pump with split casing, or a single-stage, single-flow, vertical pump would be suitable. Factors favorable to a two-stage, double-flow pump are:

   a) It would have an efficiency sufficiently high to be acceptable.

   b) It has a long-term record of proven performance within the head, capacity and specific speed range.

   c) The hydraulic thrust is balanced.

   d) Pressures on the impeller and shaft seals are less than for other type of pumps.

   On the other hand, the principal advantages of a single-stage, single-suction pump are:

   a) It is the simplest type with the least number of vulnerable parts.

   b) It can be inspected without dismantling and can be dismantled for maintenance more easily and quickly than any other type.

   c) The shaft does not project into the waterway and hence can be made as stiff as desired without effect on the impeller or volute.

   d) It is the most efficient type of pump.

   We recommend that the programs of research now under way, and such additional research as is pertinent and practicable within the time limitations, be concentrated on determining which type of pump is most suitable for a two-lift system.

3. A single-stage, single-flow, vertical pump would be optimum beyond question for a three-lift system.

EVALUATION OF ALTERNATIVE PUMP-LIFT SCHEMES

Before discussing the merits of the alternative pump lift schemes, I should like to summarize some of the most pertinent facts recently disclosed and briefly explain our philosophy in determining which alternatives we would present. In the Bechtel January 1965 report and the February 23, 1965 letter presenting specific recommendations, we stressed the advantages of two-lift and three-lift schemes utilizing single-stage pumps. We further recommended:

> "Provided that current geological investigations do not disclose any compelling deficiency that would make it infeasible, Pastoria Creek is considered to be the optimum route and should be selected for either a two-lift or a three-lift system".

Subsequent geological explorations by DWR and by Bechtel were directed primarily to the Pastoria Creek routes. As Mr. McClure has told you, some conditions were found to be more difficult than had been anticipated, others approximately as expected and some slightly more favorable. The most significant disclosure, as it affects the alternatives, concerns the middle reservoir of the Pastoria Creek two-lift system. A reservoir at approximately this same site also was considered by DWR in two of their schemes recently investigated. Unfortunately this reservoir, which was a key element of the scheme which we previously had considered to be most favorable, was found to have two major deficiencies that would make its use more expensive.

In the upper regions of this reservoir, at the site of the pumping plant for the second lift, a slide which previously had been identified was found to be much larger than had been estimated, containing some $3\frac{1}{2}$ to 4 million cubic yards. Likewise, one of the abutments at the damsite was found to contain weaker material to a greater depth than had been anticipated. While we do not consider either of these deficiencies to be fatal as regards the feasibility of providing safe structures, the measures required to overcome these geological deficiencies would increase the cost to such extent that schemes utilizing this reservoir would approach the cost of alternative two-lift schemes along the ridge location. Thus, the Pastoria Creek and the combination ridge and Pastoria Creek two-lift schemes no longer have a strong economic attraction.

The additional investigations verified the geologic suitability of the two lower reservoirs of the Pastoria Creek three-lift scheme, indicating both of these to be slightly more favorable than previously estimated. The recent explorations by both DWR and Bechtel at the upper Pastoria Creek reservoir site verified our previous estimate of both the dam and reservoir sites. We find them to be adequate but not ideal. We previously had judged the damsite to require rather expensive treatment to ensure safe design and construction. The explorations checked the necessity for treatment about as planned. Potential slide areas in the reservoir were determined to be of insufficient size to be a serious hazard.

Thus, it is Bechtel's opinion that a three-lift system up Pastoria Creek, or any other system utilizing this upper reservoir, could be constructed safely without excessive cost increases over those previously estimated. In fact, we believe that any additional more conservative measures that might be proposed within reason would not result in cost increases greater than the probable increase in cost of Tunnel No. 3 along the ridge location, which has been disclosed to have some difficult problems.

However, in evaluating complex geological and engineering problems,we recognize that there are many indeterminate factors upon which the judgement of highly qualified experts might not agree. We note that the DWR Seismic Consulting Board, in its report of April 8, 1965, concluded that ".... while the crossing can be effected by either scheme the Ridge scheme is preferable to the Canyon scheme in that it is less vulnerable to damage and presents less potential hazard to life and property."

This Board also expressed concern over the possible location of a dam and reservoir in the vicinity of the Garlock fault, which is the upper reservoir just discussed. While Bechtel agrees in general that the ridge location is geologically preferable to the Pastoria Creek location, we do not consider that any large degree of difference exists. On the other hand, it would be impracticable, if not impossible, to substantiate our views of these highly complex and intangible conditions in a completely convincing manner. We, therefore, concede that it would not be prudent to adopt a Pastoria Creek location without much more detailed geological investigations. Time does not permit such further explorations on this project.

Therefore, in the current presentation, Bechtel has limited its consideration to schemes along the ridge. Furthermore, since some concern has been expressed over the possible inadequacy of a 6 minute storage allowance in balancing tanks on multi-lift ridge schemes, we have not carried forward the two-lift and three-lift schemes utilizing balancing tanks which were presented in our January 1965 report. This does not indicate that we consider these schemes inadequate or infeasible. It must be admitted, however, that 30 minute storage capacity is preferable to 6 minute storage capacity. Furthermore, except to revise the cost estimates to reflect the recently adopted smaller system capacity, we do not believe that any significant change would be made by further consideration of the balancing tank schemes. If it is desired to give further consideration to these schemes, which appeared relatively favorable economically, it is believed that appropriate adjustments could be made readily to compare them with schemes currently being considered.

Thus, our current presentation considers only two schemes along the ridge; a single-lift scheme and a two-lift scheme utilizing an intermediate reservoir. The two-lift scheme is analyzed for both a single-stage, single-suction pump and for a two-stage, double-flow pump. As Mr. Hart has explained to you, the current cost comparison of these two schemes is based upon a total system capacity of 4100 cfs. and upon estimated pumping costs at the currently expected power rate, which is lower than previously considered.

Our analysis of these two alternative ridge schemes discloses no major difference in construction cost that would strongly favor one scheme over another. Likewise, the present worth of total costs does not vary sufficiently to be a deciding factor in selection of the system. It is Bechtel's opinion, as mentioned in my Introductory Remarks, that high dependability is the most important single factor. We believe that this consideration should completely outweigh the small cost differences indicated.

Thus, we consider the determining dependability considerations to involve:

1. The pumps and associated equipment, such as valves.

2. The discharge pipes and tunnels.

3. The structures, such as the pumping plants and any dams etc. required.

As regards the pumps, Bechtel's opinion already has been expressed quite strongly to the effect that "four-stage pumps should not be adopted under any circumstance". This, in our judgement, effectively rules out a single-lift system.

As regards discharge pipes, whether surface or underground, it is unquestionable that the headers, the complex branch connections and the main pipes would present less problems in a system designed for a maximum head of 1000 feet than for a system designed for 2000 feet. It is obvious that plate thicknesses for a two-lift system would be only one-half those of a single-lift system. This would simplify the problems of design for indeterminate stresses at connections, and the construction problems of placing and welding or otherwise connecting the units. Furthermore, pipes of a two-lift system are most economical when constructed of thoroughly proven penstock steels, thus eliminating from consideration the questionable high strength, quenched and tempered steels, such as T-1. It is therefore our opinion, from the standpoint of dependability of the discharge pipes, that it would be more prudent to adopt a two-lift system.

As regards pumping plants, dams and reservoirs and other structures appurtenant to either system, we see no preference in dependability. Both systems would utilize the same forebay and lower pumping plant site. The sites for the intermediate dam, reservoir and pumping plant on a two-lift system are relatively excellent - much the best of any of the alternative sites previously considered. It further is our opinion that a two-lift system can be operated as conveniently and as reliably as a single-lift system.

CONCLUSIONS AND RECOMMENDATIONS RE ALTERNATIVE SCHEMES

On the basis of Bechtel's investigations to date, as described in the reports previously presented and the reports and discussions presented here today, Bechtel's conclusions and recommendations concerning alternative schemes for the Tehachapi Pump Lift are as follows:

1. We recommend against adoption of a single-lift scheme, both as regards the type of pump required and as regards the large, high pressure discharge lines. It is our opinion that a single-lift system would entail the most risk in indeterminate factors in design and construction of the high-head components of the system. We believe that the four-stage pumps required would be the least dependable, the least efficient and the most expensive in operating cost of the types of pumps that have been considered for Tehachapi.

2. We recommend that a two-lift system similar to Scheme XII be adopted for the Tehachapi pump-lift system. It is our opinion that the two-lift system will be the most dependable and the least expensive in long term than any other lift system.

3. We recommend that the type of pump to be used in the two-lift system be left open for further consideration and that a strong program of research be concentrated on two-stage, double-flow pumps and on single-stage, single-flow pumps to determine which is most suitable for Tehachapi. It is not beyond the realm of possibility that a two-stage, double-flow pump might prove best for the first and highest lift and that a single-stage, single-flow pump might prove best for the second and smaller lift, where the pumping plant site is such that the higher submergence required for this type of pump would not increase the cost significantly.

4. If unforeseen conditions should be disclosed that would make it infeasible to adopt the two-lift system recommended above, Bechtel would strongly favor

selection of some other two-lift or three-lift system, using balancing tanks if necessary, rather than to adopt a single-lift system.

M. L. Dickinson

M.L. Dickinson
Chief Hydraulic Engineer

MLD/ds

TEHACHAPI PUMP LIFT SYSTEM

ADDITIONAL REMARKS ON EUROPEAN PUMPING PRACTICES
referring to Second Report, January 1965

by
Professor Hans Gerber, Swiss Federal Institute of Technology, Zurich, Switzerland, Consultant to Bechtel
May 1965

1. INTRODUCTION

Just before leaving Zurich to attend the Meeting with the Tehachapi Crossing Consulting Board, I had the opportunity to look through with Dr. Sutherland the report we worked out together and I have been commissioned by him to represent the statements and considerations of this report. In addition to this Second Report, which has been in your hands for some months, we have taken into consideration all facts developed in the meantime, including the model testing program, on which Professor Neale will report separately. We are fully aware of the necessity to make some major decisions in the near future and Dr. Sutherland and I would be happy if our investigations on the European Pumping Practices would contribute in some manner to this important and difficult task.

In the following I will review the important questions in the same order that we think they should be dealt with and should be taken into consideration in the final decisions.

2. CHOICE OF LIFT SYSTEM

We think that the choice of a single-lift, a two-lift or a three-lift solution is first of all a topographical, geological and seismic problem. It should clearly be stated that, independent of costs, it would be possible for all three lift solutions to have reliable and rugged pumps built, and for all these pumps long years of experience of different kinds are available.

In the following I want to emphasize the advantages and disadvantages of the different lift solutions, as it seems to come out from our European experience, which

I agree for the moment, may differ in some aspects from the American one.

3. SINGLE-LIFT SOLUTION

The only advantage of the single-lift solution seems to be the simplicity of the whole scheme for the future operation, with a minimum of electrical installations and a minimum of plant staff.

On the other hand, there are several rather important disadvantages in existence.

3.1 First of all, I do not think that for a four-stage pump peak efficiencies will be possible. No multi-stage, single-flow prototype pump has been built up to date for such a high specific speed which is necessary for an optimum efficiency. This conviction has not been altered by any model test result, as far as I know.

3.2 If, as it has been discussed, no T-1 steel will be used for the penstocks or tunnel linings, I think that the piping system must become substantially complicated and expensive.

3.3 Furthermore, I think that for the future maintenance and replacement work of seal rings, for example, the four-stage pump would be by far the most disadvantageous.

4. THREE-LIFT SOLUTION

Of course, the big advantage of this solution would be the fact that it could be considered as an almost classical one, specially in reference to the experiences on pump design so far available in this country. From the viewpoint of American experiences it is absolutely understandable that those who will have to run the scheme afterwards would no doubt prefer a three-lift solution. I think that this remark would refer especially to the good experiences of the Metropolitan Water District of Southern California with the pumping stations of the Colorado River Aqueduct.

However, I feel that this solution would be conservative to an unnecessarily high degree, and thus lead to the following disadvantages.

4.1 The costs would probably be relatively high, both for the civil engineering section and the mechanical and electrical equipment.

4.2 The operation of three pumping plants would not simplify maintenance and operation.

5. TWO-LIFT SOLUTION

Taking into account all these facts and experiences on one hand and the severe operating conditions on the other hand, I have had the conviction from the beginning of my participation on the studies for the Tehachapi Crossing, that the two-lift solution would be the right intermediate one. Furthermore, it is certain that the hydroelectric equipment can be chosen in a satisfactory manner.

But before entering in detail into these questions, may I say that those who have finally to decide on the solution to be adopted must clearly decide first whether they are willing to make an extrapolation beyond the existing field of experiences of important pumping plants. Two types of pumps are in the foreground. I will discuss each type in some detail.

5.1 Single-stage, Single-suction Pumps.

Taking into account all the efforts which in the last few years have been made in the field of the reversible machines, both in the United States and Europe, this type of pump would follow the same general line of development. This type is no doubt of the simplest design, would be well appropriate for a vertical shaft solution, promises a relatively high efficiency and can easily be designed for a rugged and reliable service. It seems to be right, therefore, that this type of pump, well known especially in the United States, appears to be the first type to be considered for a two-lift solution, provided one is aware and ready to accept the following disadvantages:

5.1.1 The use of a single-stage pump of this size and this specific speed requires an extrapolation beyond existing experience. I do not think, for example, that the Taum Sauk scheme is an extremely convincing example to prove that this type can be designed and put in satisfactory operation immediately and without troubles.

5.1.2 Furthermore, as a man who has helped to design and who has put in operation personally quite many huge hydraulic machines, turbines and pumps, I am not ready for the moment to accept as proof all those reversible machines which are today under construction or in erection, such as Cruachan, Roenckhausen or Robiei. There is a fair chance that they will work out well, and as engineers we all expect and hope that there will be success in all plants, but once more it is my feeling that it is not yet proved.

5.1.3 There is not doubt that this solution will need by far the highest submergence to avoid under all circumstances cavitation troubles, noise, erosion and efficiency drop in the future. During the Symposium on Pump Design and Operation which has been held at East Kilbride in Scotland three weeks ago, it was clearly shown how difficult this question is and how careful and with caution it must be handled.

5.1.4 In this connection, the use of surge tanks, instead of a free intermediate reservoir could easily provide sufficiently high submergence. But I do not think that the surge tank solution would lead to the same degree of stability and flexibility in operation as it would be with a reasonably well dimensioned intermediate reservoir.

5.2 Double-stage, Double-suction Pumps.

For all these questions, in my opinion, this type of pump, which has been developed to a high standard in Europe for more than 30 years, would answer in the very best manner. The following facts may be mentioned:

5.2.1 There would not be necessary any extrapolation, neither for head nor for size, power or speed. All the Tehachapi conditions are covered with the experiences of about 68 different pumps in almost 30 plants.

5.2.2 The hydraulic axial thrust is practically fully balanced. I know that for a vertical shaft solution, as provided for Tehachapi, this is of no great importance, but I shall come to this question later.

5.2.3 The efficiencies have proved to be reasonably high, both on models and on the prototypes, and they are still suitable to be increased by going to a somewhat higher specific speed.

5.2.4 This type of pump can be designed exactly for the same reliability and the same ruggedness as a single-stage, single-suction pump, especially due to the fact that the head per stage is half the value.

5.2.5 The necessary submergence for good cavitation conditions will be considerably smaller than for the single-stage type, even if we remember that because of the shaft going through the entrance bends some additional submergence would be necessary.

As before, the disadvantages of a two-stage solution also should be mentioned. These are namely two of them:

5.2.6 This type of pump is more expensive in first cost. However, depending on the civil engineering conditions, this difference in cost could be reduced somewhat, or perhaps compensated by the smaller submergence required.

5.2.7 It is clear that the shafts through the entrance bends reduce the efficiency in a well known manner. On the other hand, this disadvantage is clearly compensated by the elimination of one disc friction of the impeller hub.

6. In order to be as complete as possible, I would like to mention that there is also a third type of pump which should not be neglected in the overall considerations and this is the <u>single-suction</u>, <u>double-stage type</u>, with the impeller overhung. There are just 2 pumps of this type in operation, namely in the Ferrera Plant in Switzerland, with horizontal shaft, for 1560 feet, about 40,000 HP. We have just finished the computation of the acceptance tests,

which I carried out last year as a neutral consultant. If you put the efficiency of 88.4% for a specific speed of 1340 only in Fig. No. 8 of our Second Report, it can be seen that this type of pump is exactly placed on the upper limit of the band we have established for single-suction pumps.

These pumps are running absolutely quiet. No overhaul work has been made before the tests, with about 2500 working hours, and not always clean water to be pumped. If the elimination of a shaft through the entrance bend is proved to be advantageous both for efficiency and for cavitation, this type has these advantages for the first stage at least. For the second stage there is no cavitation question.

This type of pump could easily be designed for a much higher specific speed, with much increased efficiency, especially when a vertical shaft position is provided.

7. SHAFT POSITION

I know that for the Tehachapi plant vertical shaft position has been provided. This solution will surely be the right one for single-suction pumps, both single-stage or even double-stage, overhung.

If the double-suction, double-stage pump type is to be considered, then both positions of shaft have been chosen; a few only with vertical shaft, namely Oberaar with a natural high submergence, Ffestioniog and Villa Gargnano, where in both plants the upper runner would better have the position of the lower one.

8. MAINTENANCE AND OPERATION

Generally speaking, it is considered that for operation the horizontal shaft solution brings some advantages. I would agree with this general opinion as the whole unit is on the same floor. On the other hand, vertical shaft units do not require as much floor space as those with horizontal shaft. But even in underground stations, quite often horizontal shaft machines can be seen.

Especially for double-suction pumps, the advantage of horizontal shaft position is generally considered to be important.

Referring to this special type of pump, I must mention the horizontal splitting of the housings, where the sealing of the two parts seems to be resolved. Wherever these split housings are adopted - in Bringhausen, Provvidenza, Limberg, and now in Vianden - the plant operating staff is convinced of the many important advantages of this solution.

I personally am convinced that with all shaft positions and types of pumps suitable for the Tehachapi project, the horizontal splitting of housings would lead to the best conditions for maintenance, overhaul and repair work, especially also for the exchange of seal rings.

9. SEAL RINGS AND HEAD PER STAGE

I think that there is today no problem for a good seal ring solution up to 1000 feet per stage. In Europe many high-head single-stage Francis units up to 1500 feet are running successfully for many years. From this side I do not see any objection against the adoption of single-stage pumps in a two-lift solution.

Naturally the materials for the seal rings must be chosen in the right manner, but this has proved to be possible.

10. MODEL TESTING

The question of model testing will become much easier and the corresponding program can be adjusted in a very efficient manner as soon as the question of the lift system is settled, which as it seems, will be the case in the not too distant future.

But then it will be of the highest importance to allow sufficient time to carry to a good end and to a sufficiently broad extent all the model research work now under way. This remark applies equally to the DWR program and to the Bechtel program now under way at the NEL, about which Professor L. Neale will give a special report.

The most important thing seems to me to be the testing of as many models as possible under the full Tehachapi head. Even then the interpretation of the test results must be made very carefully and in a complete manner, especially those dealing with the final setting of the pumps.

11. FINAL CONCLUSIONS AND RECOMMENDATIONS

Taking into account the foregoing reflections, I would like to express my personal opinion on the Tehachapi problems as follows:

11.1 If there are no imperative reasons from the civil engineering and geological side, or from heavy differences in the first cost, the two-lift solution seems to be the best in all aspects.

11.2 If this solution is adopted, there are different types of pumps available. It is certain that the single-stage, single-suction type has a fair chance to become a good solution, but that can be proved by facts perhaps in a few years when the plants now under construction are running.

11.3 If for a two-lift solution, the overall risks on the pump side have to be reduced to the lowest possible minimum, then the double-stage, double-suction pump type is by far the best on the basis of long years of experience and can be built without any extrapolations.

11.4 All other questions - shaft position, seal rings, etc. - will be a consequence of the chosen type.

11.5 The model testing program is of the highest importance and value and should be completed and if necessary extended so that a maximum of information will be made available. Sufficient time will be the first condition for that purpose.

H. Gerber.

Hans Gerber
Consulting Engineer

San Francisco, ~~March~~ MAY 3, 1965
HG/ds

SUMMARY ON REPORT OF SURVEY

of

AMERICAN PRACTICE IN LARGE CAPACITY
HIGH HEAD CENTRIFUGAL PUMPS AND PUMP-TURBINES

and its relation to studies
for the

TEHACHAPI CROSSING PUMPING PLANT

of the

CALIFORNIA STATE WATER PROJECT

for

THE METROPOLITAN WATER DISTRICT
OF SOUTHERN CALIFORNIA

prepared by

RAY S. QUICK, CONSULTANT

BECHTEL CORPORATION
SAN FRANCISCO

MAY 1965

SUMMARY

The American Practice Report on Large Capacity High Head Centrifugal Pumps is based on a survey undertaken in connection with the studies of pumps adaptable to the Tehachapi requirements. Sixteen of the major plants were studied. All are of the vertical, single suction, single stage design, including recent pump-turbines now adapted to heads in excess of 1000 feet.

The simplicity and ease of maintenance of the vertical Francis turbine had been established for many years, including long experience in the selection of suitable materials for runners, seals, casings, valves and other important items so it was deemed advantageous to apply this background to a modern high capacity pump providing that suitable efficiency and operating characteristics could be developed.

When The Metropolitan Water District of Southern California was faced with the selection of the best type of pump for the Colorado River Aqueduct, about 1930, they first investigated pump models of conventional design, then later decided to check on the single suction, single stage type, due to its greater simplicity and more convenient station piping. Arrangements were made to conduct a series of tests at The California Institute of Technology, under the direction of the late Dr. R. T. Knapp. The single suction, single stage designs offered by several manufacturers proved to have excellent efficiencies of 90 percent or higher, with stable characteristics, resulting in the selection of this type for all stations. Each of five pumping plants was equipped initially with three pumps, followed by six more, in later years. These plants have been running continuously, with short time shutdowns for inspection and maintenance, with excellent results. This experience is the real start of the modern single stage, single suction type.

The Grand Coulee pumping plant of the Columbia Basin project of the United States Bureau of Reclamation was being studied about the same time as the Colorado River Aqueduct was being completed. It was decided to take advantage of the development work done at The California Institue of Technology and to supplement it with model studies of pumps best suited to the variable head conditions at Grand Coulee. The motors were to be of a record size at that time, namely 65,000 horsepower, so a special method of starting, using one of the main hydroelectric units at Grand Coulee, at reduced speed, was devised. The model efficiency exceeded 90 percent and the field tests showed 93.9 percent, a record value. Grand Coulee also is of the single suction, single stage vertical design.

When the reversible pump-turbine was developed for peak power use by public utilities, it was most convenient to follow existing practice by investigating the possibilities of the vertical, single suction, single stage design. Here there is a dual operating problem to provide a single unit which will have satisfactory operating characteristics when operating either as a turbine developing power, or as a pump delivering water at a comparable electrical load against the full head. Fortunately, the pioneering work at The California Institute of Technology for the Colorado River Aqueduct model pumps had established the full characteristics of the unit when operating in either direction so it became evident that a pump was also an efficient turbine. Wicket gates or guide vanes were added for load regulation as a turbine and later model tests were undertaken by various manufacturers to establish the characteristics which could be realized with a dual purpose unit. This soon led to the modern design of which a number in this field are included in the January 1965 report. Here the range in heads is far more extensive than has been reached to date in large single suction, single stage pumps. Pump-turbines are operating successfully under heads up to nearly 900 feet. Units are under construction for installation under heads up to over 1000 feet in the United States and 1200 feet in Scotland. The latter plant, known as Cruachan, will contain four units of 100 megawatts each, two being furnished by Boving, and two by English Electric-Sulzer. This plant is nearing completion and expects to initiate service in 1965.

All of the above mentioned pump-turbines are of the single suction,single stage design. Submergence has been selected with due regard to elevation and model tests. Mechanical design is based on hydraulic turbine practice.

Taum Sauk was placed in operation in 1963, and aside from minor difficulties experienced at the time of start-up, has been operating successfully. The initial difficulties consisted of a resonant vertical vibration of the rotating assembly, in combination with a water-hammer vibration in the drain piping used for depressing the water to below the runner-impeller on start-up, which was corrected by changing the drain piping, and an increasing hydraulic uplift on the impeller when running in overspeed as a free running turbine. This latter tendency was overcome by applying additional pressure above the impeller, under overspeed conditions and will be corrected in later designs by a change in configuration of the lower cover.

APPLICATION OF AMERICAN PRACTICE TO TEHACHAPI

American practice in large capacity centrifugal pumps and pump-turbines indicates that the single suction, single stage pump, with a vertical setting, can be applied successfully to either the three lift or the two lift schemes for Tehachapi.

In the three lift scheme, with about 654 feet of head, existing developments would indicate a high efficiency and conservative design, of very simple configuration. Long life and minimum maintenance could be expected.

In the two lift scheme, with about 982 feet of head, a similiar design as for the three lift scheme could be employed, with due regard to submergence. Comparable efficiencies could be expected.

In both of the above types, hydraulic turbine practice is available to establish adequate criteria for strength of casing, covers, main shaft, guide bearing and other principal parts. Impeller seals on pump-turbines follow turbine practice for which there is much information for heads of 1000 feet, or more. Seals should be lubricated with clean water, not only to prolong life, but also to keep the seals cool and minimize any tendency for seizure when run without water in the pump casing.

A vertical setting simplifies the selection and arrangement of the thrust bearing, which can be combined with the driving motor; the use of high pressure oil for minimizing friction for starting; the handling of the rotating elements by means of the station crane, and minimum floor space. It also permits setting the inlet of the impeller at minimum elevation, or maximum submergence, with minimum excavation, and permits embedding the casing in the concrete foundations which provides maximum rigidity.

Model tests of homologous model pumps, tested at prototype head, will establish performance and cavitation data which will be required for final design. This information should include data on axial and side thrust, for determining thrust bearing loads, uplift under overspeed as a turbine, and main shaft deflection and its relation to seal clearance under normal and abnormal operating conditions.

American practice, as covered by the report of January 1965,does not cover any designs which would apply to the single lift scheme. However, it does offer a very promising solution for the two and three lift plans.

American practice in starting of heavy duty motor driven pumps offers a choice which will be determined by electrical limitations and economics at Tehachapi. This is more fully covered in the Report on American Practice.

Discharge valves, as covered in the Report on American practice, would favor the spherical type for Tehachapi. Such valves are used under heads exceeding 2000 feet in hydroelectric installations, so would offer no problem at 1000 feet head.

Respectfully submitted,

Ray S. Quick

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Bechtel Corporation

MODEL PUMP TEST PROGRAM
TEHACHAPI CROSSING

CALIFORNIA STATE WATER PROJECT
METROPOLITAN WATER DISTRICT
OF SOUTHERN CALIFORNIA

NATIONAL ENGINEERING LABORATORY
EAST KILBRIDE, SCOTLAND

BECHTEL CORPORATION
SAN FRANCISCO, CALIFORNIA

APRIL 1965

Professor Lawrence C. Neale

PART 1. PRESENT TEST PROGRAM

Introduction

A series of model studies was initiated in September 1964 as a part of the Bechtel study of the Tehachapi Pump Lift. The purpose of these studies was to develop information that would be needed in the selection of the pump type and pump characteristics of the machines to be used for the pump lift.

The model studies were set up to provide the following:

1. A direct comparison in a single laboratory of results from each of the pump manufacturer's laboratories.
2. A relationship between efficiency and specific speed for the model pumps.
3. A comparison of single stage and two-stage pumps.
4. Correlation of other pertinent data as might be generated from the tests.

The program at present stands with six pumps tested and the seventh under test.

Laboratory

In selecting a laboratory for the studies, a survey was conducted of the available independent laboratories throughout the world. As a result the National Engineering Laboratory in East Kilbride in Scotland was chosen. The NEL enjoys the confidence of manufacturers of pumps and hydraulic turbines throughout the world. This confidence is the result of the combination of an excellent staff with wide experience in the pump testing and design field, and a fine installation of testing facilities that are unequalled in the world today. A detailed description of the laboratory facilities is given in NEL report entitled "Progress Report No. 2", dated January 1965. For the record, this description is contained in Appendix A of this review.

Testing Program

In the initial program it was believed that because of time restrictions the pump manufacturers could not produce models specifically designed for the Tehachapi Crossing. Therefore a number of manufacturers were asked to submit lists of existing pump models which they would offer to the program. These models were to be roughly applicable to the problem and the test results would contribute to the objectives listed above. Eighteen models were offered by six different manufacturers and from this group seven models were selected with each manufacturer represented in the selections. Three pump models were of the 2-stage single entry type and four pump models were single stage, single entry type, while the specific speed varied from 105 to 150 in the metric system. (1480-2130 U.S.).

The detailed testing procedures are described in the NEL Report entitled, "Progress Report No. 2", Interim Report on Tehachapi Pump Test Programme, dated January 1965. This procedure is also reproduced in this report as Appendix B.

Results

The test results on five of these models completed indicate that the efficiency at "best efficiency point"(b.e.p.) as determined at NEL is in agreement within 0.5% of the manufacturers' furnished data. On the sixth model, the comparison is difficult at this time since the manufacturer wishes to check his model immediately on its return to his laboratory. Results for tests conducted at other than "b.e.p." show variations of as much as 2% from the manufacturers curves and data.

It should also be pointed out that the pressure tappings of the manufacturer have shown variations of more than 0.5% from the standard NEL tappings used in each test.

These results therefore indicate that the test work done in most laboratories is of high quality, comparable to that performed at NEL. However, in order to get a comparison as important as that for the Tehachapi Crossing, tests in a single independent laboratory give valuable data upon which to base the decision.

Data for these tests is presented in a non-dimensionalized form in Progress Reports Nos. 2 and 3 entitled, "Interim Report on Tehachapi Pump Test Programme" dated January 1965 and April 1965. Figure 14 in Report #2 and Figure 4 in Report #3 give the specific data as differences in efficiency versus flow. In viewing these plots it should be kept in mind that for Pump D, the pump model was operated at a reduced speed and the results are reflecting this reduced speed.

In a number of the models the construction is such that a labyrinth is placed in the model and leakage through this arrangement is used to balance the thrust on the impeller. In evaluating test results the treatment of this leakage must be consistent and the details spelled out in each case. Variations in computed efficiency due to different treatment of this leakage amounted to a spread of 3-1/2% in efficiency for one pump model in the present series.

The relationship between specific speed and efficiency is shown graphically as Fig. 6 in Progress Report No. 3 noted above. This plot shows that the efficiency increases with the increase in specific speeds. It also appears from the data that an increment of efficiency exists in the case of single stage model pumps over 2-stage model pumps. Although the data from only six pump models is not conclusive, the trend is in agreement with field experience as spelled out by Gerber and Sutherland in "Second Report on European Pump Practice" dated January 1965, Fig. 10.

$S = \frac{\text{NEL TEST SPEED} \times 100}{\text{MANUFACTURER'S TEST SPEED}}$

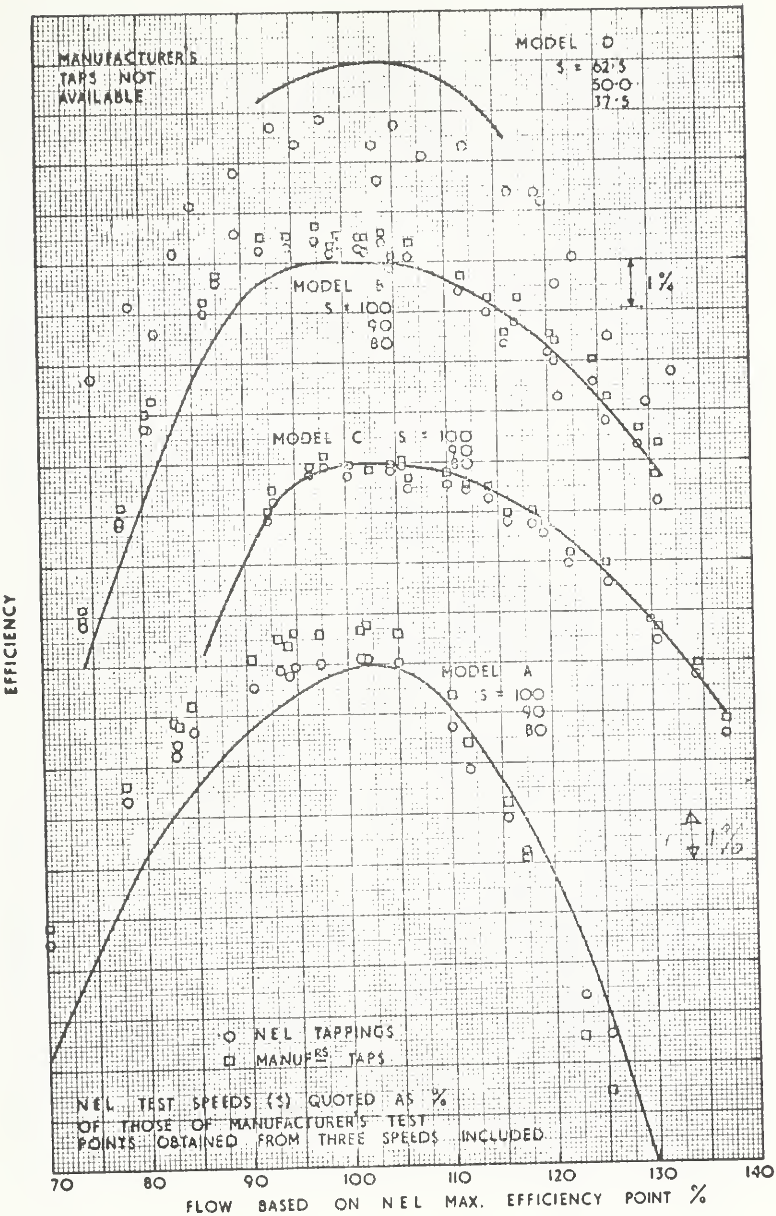

FIG 14 COMPARISON OF N.E.L. RESULTS AND MANUFACTURER'S CURVES

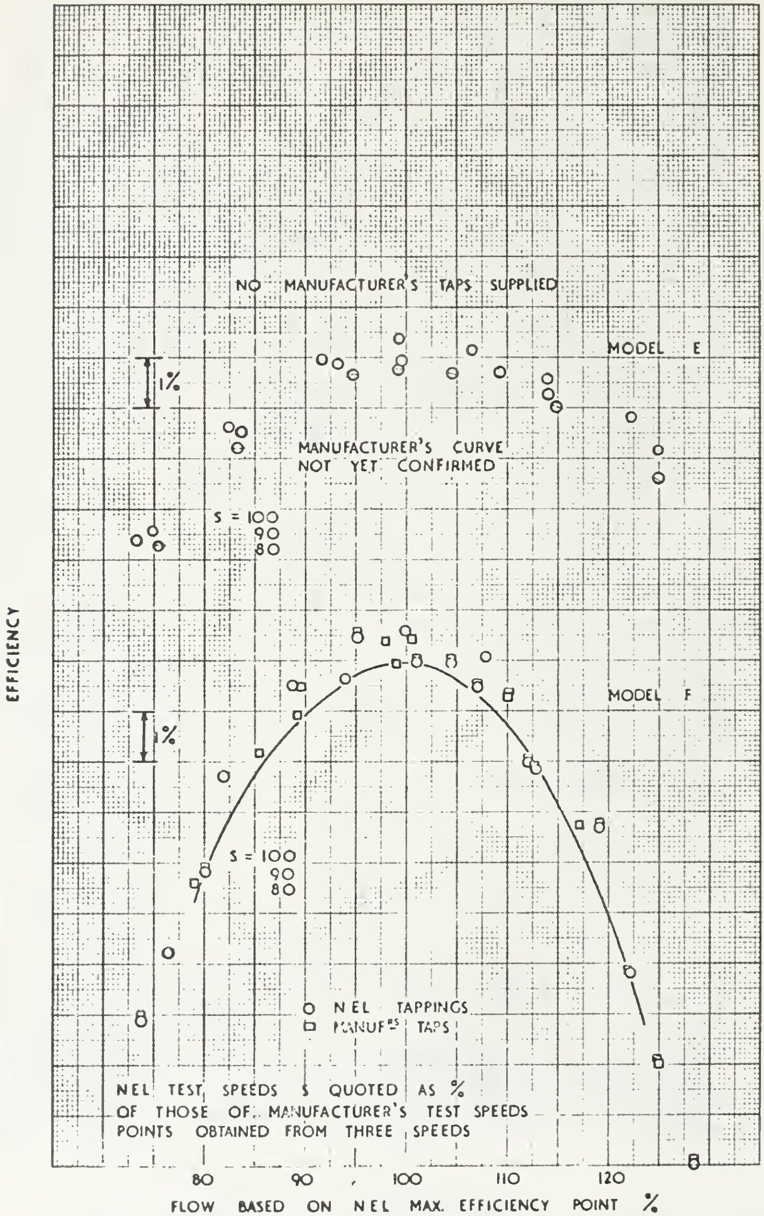

FIG 4 COMPARISON OF N.E.L. RESULTS & MANUFACTURER'S CURVES

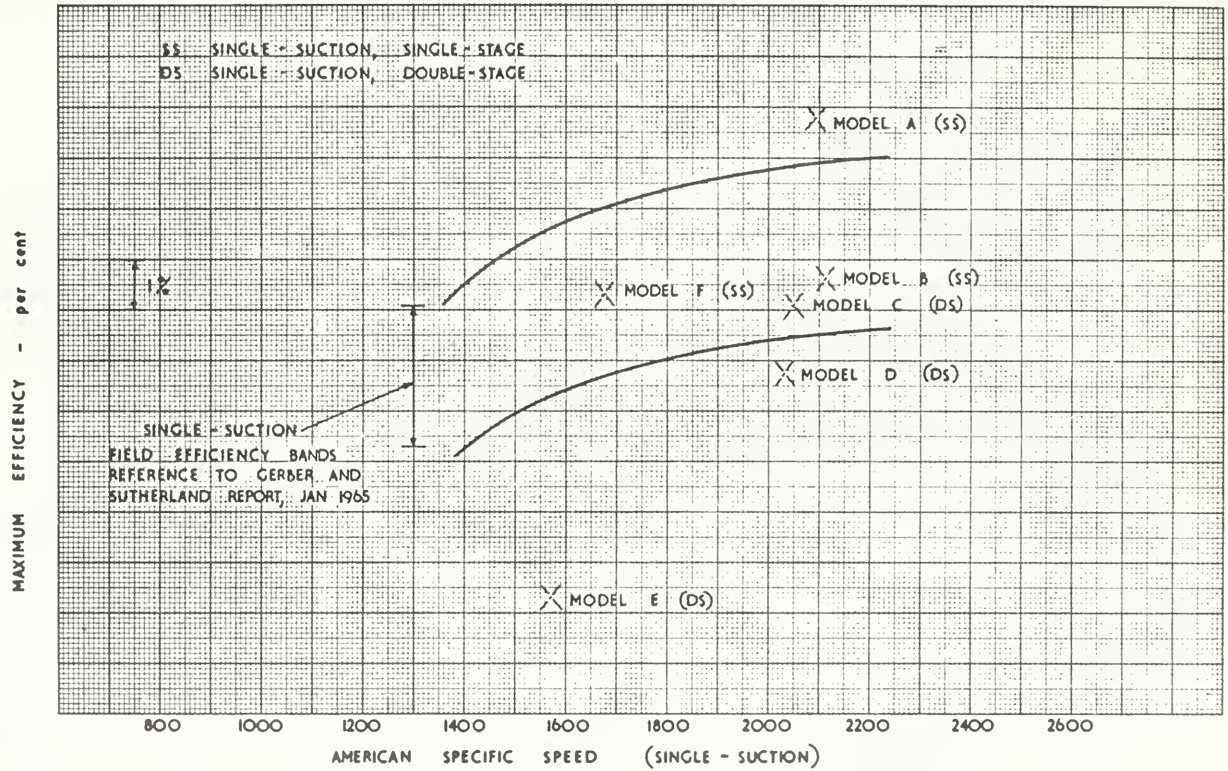

FIG 6 (REVISED) VARIATION OF MAXIMUM EFFICIENCY WITH SPECIFIC SPEED

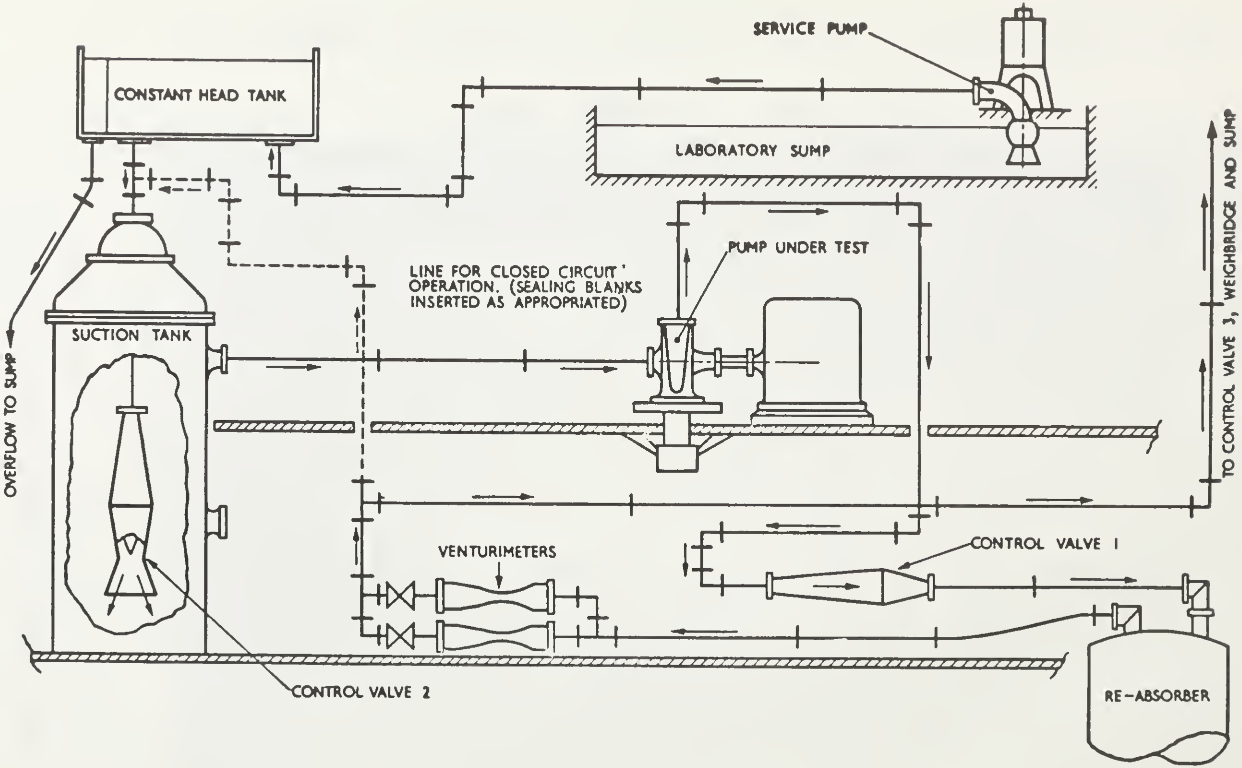

FIG 10 CLOSED CIRCUIT FOR CAVITATION TESTS

In evaluating the data from the pump tests it is pointed out again that the models are each operated at three different speeds. It is apparent that on some of the models the best efficiency point has had practically the same value for each speed. In other cases there is a considerable change in efficiency. Figure 5 in Progress Report #3 shows graphically the variations in efficiency with speed. (This figure is also presented at the back of this review). This variation is an important result of these tests since a number of pump models in manufacturers' laboratories cannot be tested at full rated prototype head and corresponding speed. If this test requirement is not fulfilled, it is necessary to compute these results at the desired conditions based on an assumption of a constant efficiency or on emperical relationships. Further comment on this particular feature will be included in the discussion of the high power test rig. later in the presentation.

The test program to date has involved cavitation studies on only one model. These studies were concerned only with determining the change in performance of the pump with a reduced "sigma" value. No attempt was made to observe cavitation developing on the pump impeller. The values obtained on the one model corresponded with those supplied by the pump manufacturer.

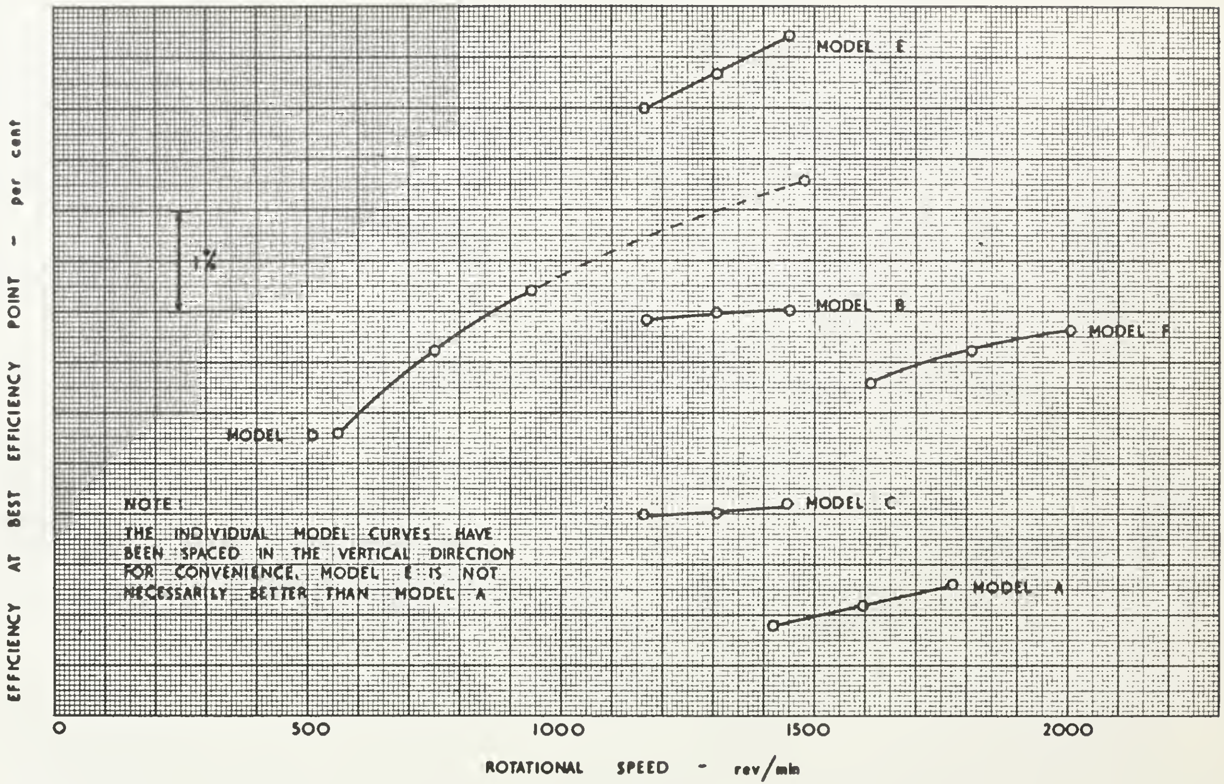

FIG 5 (REVISED) VARIATION OF EFFICIENCY WITH CHANGE IN ROTATIONAL SPEED

Summary of Present Program

1. The present program has shown a good correlation with the manufacturers' laboratories. Results at best efficiency point are within 1/2 percent.

2. Leakage treatment in each model is particularly important.

3. Test data away from "b.e.p." does show variations up to 2% from manufacturer's curves.

4. There is variation in results depending on which set of pressure tappings are taken for head measurements.

5. A definite relationship between specific speed and efficiency is apparent.

6. Some models are speed sensitive and require careful analysis if tests are not possible over complete speed range.

Part 2 HIGH POWER TEST RIG

Introduction

As the test program at the NEL developed, it became apparent that the DWR models would become available for test at NEL after testing by the individual manufacturers. In order to make these tests meaningful the test facilities at NEL were re-examined and plans initiated to expand the capability of NEL. Under these plans it would be possible to test the model pump designed to fit a single lift, 2-lift or 3-lift system for the Tehachapi Crossing.

Test Set-Up

The high power facility incorporated a 4000 HP drive and multiple speed gear box with the hydraulic facilities covering the pump models to fit the conditions listed above. The details of this rest facility are indicated in Fluids Memo No. 224, entitled, "The High Power Pump Test Rig at NEL" dated January 1965. At present some of the equipment is on the site, some in transit and some being completed.

Test Program

In order to fill out the coverage of model pump characteristics, the MWD-Bechtel program has commissioned two manufacturers to produce models to fit the two-lift concept of the crossing. These two models are single stage, single inlet models. One model is ready for shipment to East Kilbride while the other will be available in September 1965. In addition a high specific speed (N_S = 164) single stage, single suction laboratory model is to be available for test. The details of the procedure required for test on the high power test stand are presented in Fluids Memo 224.

Part 2

Summary

As indicated above the high power test rig will provide a capability of testing at the rated head on any model pump available to the Tehachapi studies. As a result it should be possible to eliminote questions posed by tests at reduced speeds. Also the cavitation characteristics can be observed and studied under the full head conditions. These two areas are most important to the overall conclusions based on the model test program.

APPENDIX A.

TEST APPARATUS

It was agreed that for all pump models submitted for test at NEL the basic test circuit should be common. A modification was made to the pipework to convert the closed-circuit pump facility, as described in reference 1, into a controlled open-circuit arrangement. Without moving the pump, model performance tests on open circuit can be made using the Laboratory's flow measurement standard or the circuit can be switched to be a closed loop so that cavitation tests can be carried out.

The circuit, shown diagrammatically in Fig. 1, consists of the pump to be tested connected between two reservoirs having exposed free surfaces. Thus water is supplied to a constant-head tank by a separate variable-speed service pump drawing from the main reservoir or sump. The model pump draws water from this tank through a second large closed settling tank. This suction tank, situated immediately below the constant-head tank, contains baffles to ensure a good velocity distribution into the suction pipe to the model pump.

The discharge from the model pump passes first through a main pressure breakdown valve and for cavitation testing would then be passed through the 12 ft. diameter reabsorber buried vertically below basement level. For normal performance testing, the discharge from the breakdown valve goes directly to the venturi meter bank and then on to a spear valve above the NEL diverter and weigh-bridge system. From there the water is returned to the main sump.

The suction pressure at the inlet branch to the model pump can be controlled by the special breakdown valve in the suction tank. Any desired value below 64 ft. absolute (+19.5 m) down to the vapor pressure can be set and held. When cavitation tests are made the reabsorber ensures that the air bubbles left after the collapse of cavitation are re-dissolved into solution in the water before it returns to the inlet of the model pump.

Straight lengths of pipe, similar in diameter to the inlet and outlet diameters of the model pump, are installed immediately upstream and downstream of the model under test, Fig. 2. This ensures that the flow pattern into the pump is the same for all tests. The pipes for all pump tests are of the same material and manufactured in a similar manner to ensure consistency throughout the tests. Measurements of the internal surface (or surface texture) of these pipes are being taken to check this.

The model pumps are all driven by the NEL No. 3 precision dynamometer, Figs. 3, 4, which is essentially a speed-controlled d.c. motor whose carcase is suspended on hydrostatic bearings using oil pressure at 600 lb/in^2. There is an automatic centralizing device and the torque is measured directly with dead weights on the torque arm and a hydraulic manometer vernier. The speed is automatically kept within ± 1/3 rev/min by a Boller and Chivens control unit, Fig. 5.

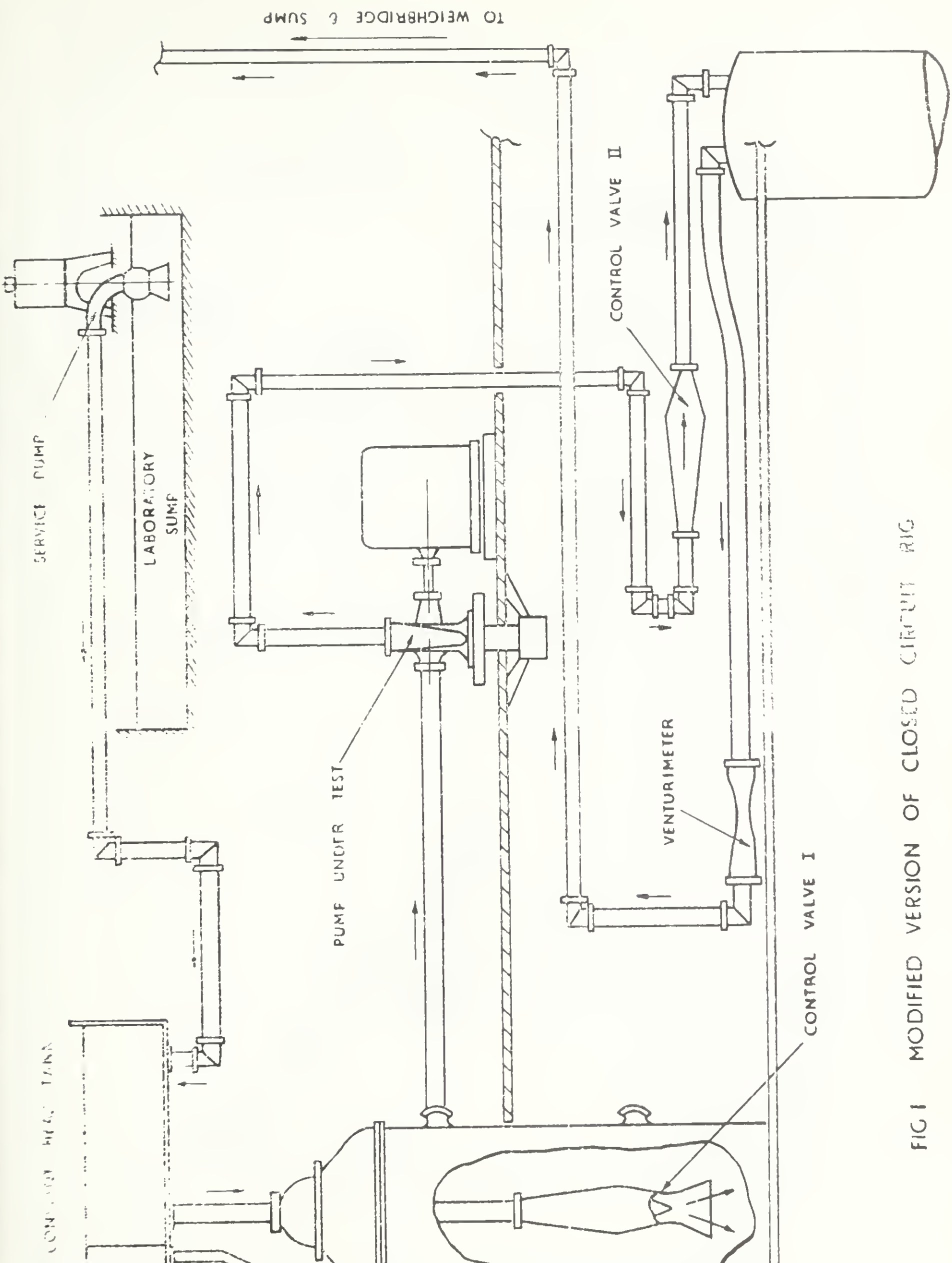

FIG I MODIFIED VERSION OF CLOSED CIRCUIT RIG

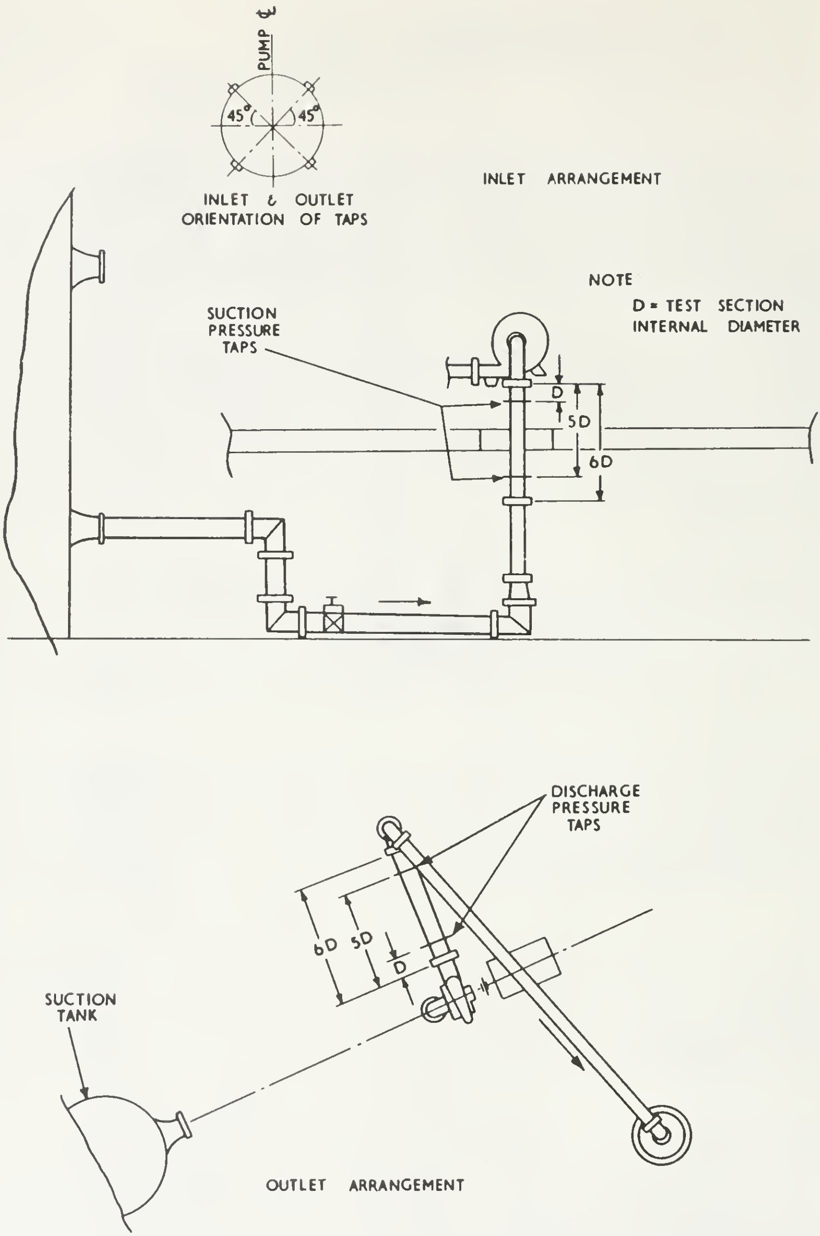

FIG 2 ARRANGEMENT OF PIPING ADJACENT TO PUMP

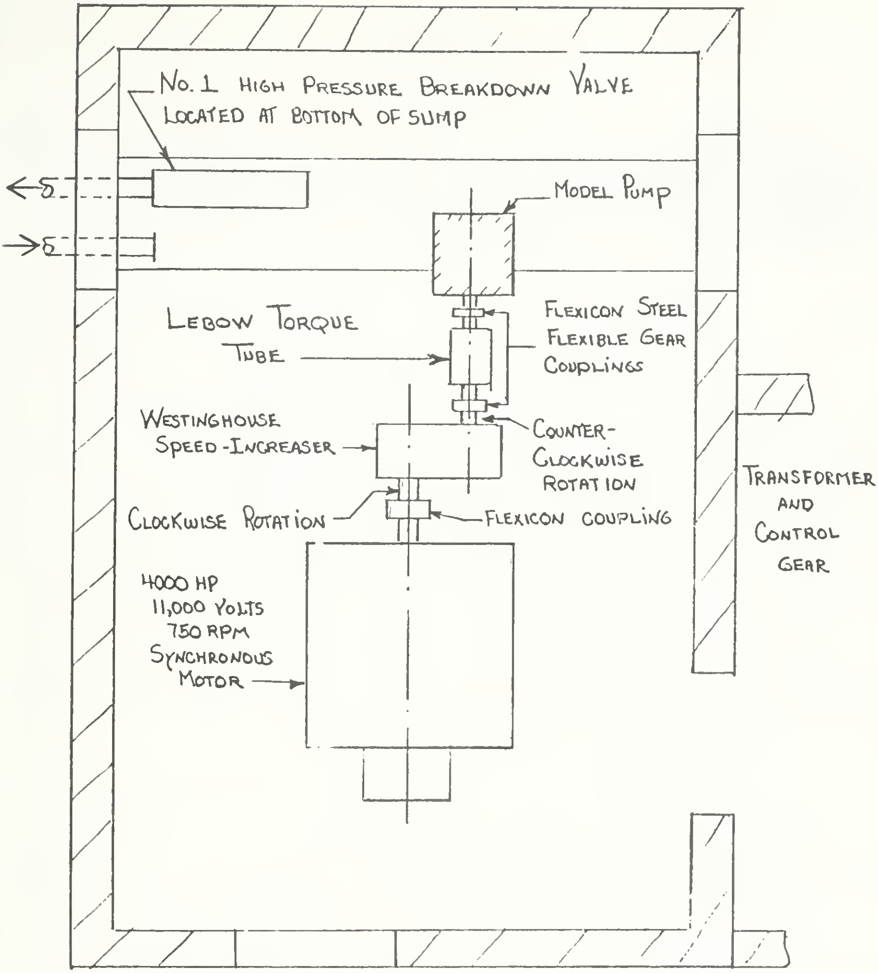

Fig. 3 4000 HP Test Facility at NEL

The dynamometer can supply, at a continuous rating, 200 hp over the speed range from 1250 to 3000 rev/min and can give a continuous power rating equal to 0.16 N for speeds from 1250 down to 500 rev/min. The output can be increased under short-time overload conditions to give a maximum torque of 1000 lb ft, equivalent to 0.19 N.

APPENDIX B.

TEST PROCEDURE

The normal method of testing pumps in use at NEL is that given in the revision of the British Standard Test Code (2) (to be published shortly). For these tests other test codes have been consulted and in addition extra measurements have been made to provide checks on the main measurements or to obtain extra information.

Head Measurements

Measuring sections

Separate sections of parallel straight pipe, six bore diameters in length, are attached to both the inlet and discharge branches of each pump model. These are shown diagrammatically in Fig. 2. A photograph, Fig. 6, shows the suction pipe. Two sets of four static-pressure holes, 1/8 inch in diameter, spaced on radial planes at 90° intervals and at linear positions one and five bore diameters from the model branch flanges are being used throughout the tests. The head readings of each individual tapping point are noted and the mean head at any particular plane taken as the arithmetic mean of the four readings.

Supplementary to the readings at the tappings in the measuring sections described above, the original manufacturer's pressure tapping points are used when available to derive a direct comparison with his measurements of the differential head.

Manometers and gauges

For head readings across the manufacturers' tapping points, a vertical U-tube mercury/water manometer, 27 ft. in height, was constructed and permanently connected to piezometric rings on the suction and discharge branches. Differential heads of up to 340 ft can be measured directly with this manometer.

For measurements of head from the NEL pipe sections, standard NEL 80-inch single limb mercury/water manometers were used, Fig. 7. On the discharge side of the model four of these were connected in series to cover the head range but provision was made when making lower head readings to by-pass those not required.

The low-pressure side of both inlet and discharge manometer systems was permanently connected to a common reservoir having an exposed water surface at the same height as the pump centerline. This level was taken as datum for all the test pressures.

For monitoring and for checking purposes, calibrated pressure gauges were included in the suction and discharge pressure leads. These were arranged at datum level to eliminate the necessity for elevation correction.

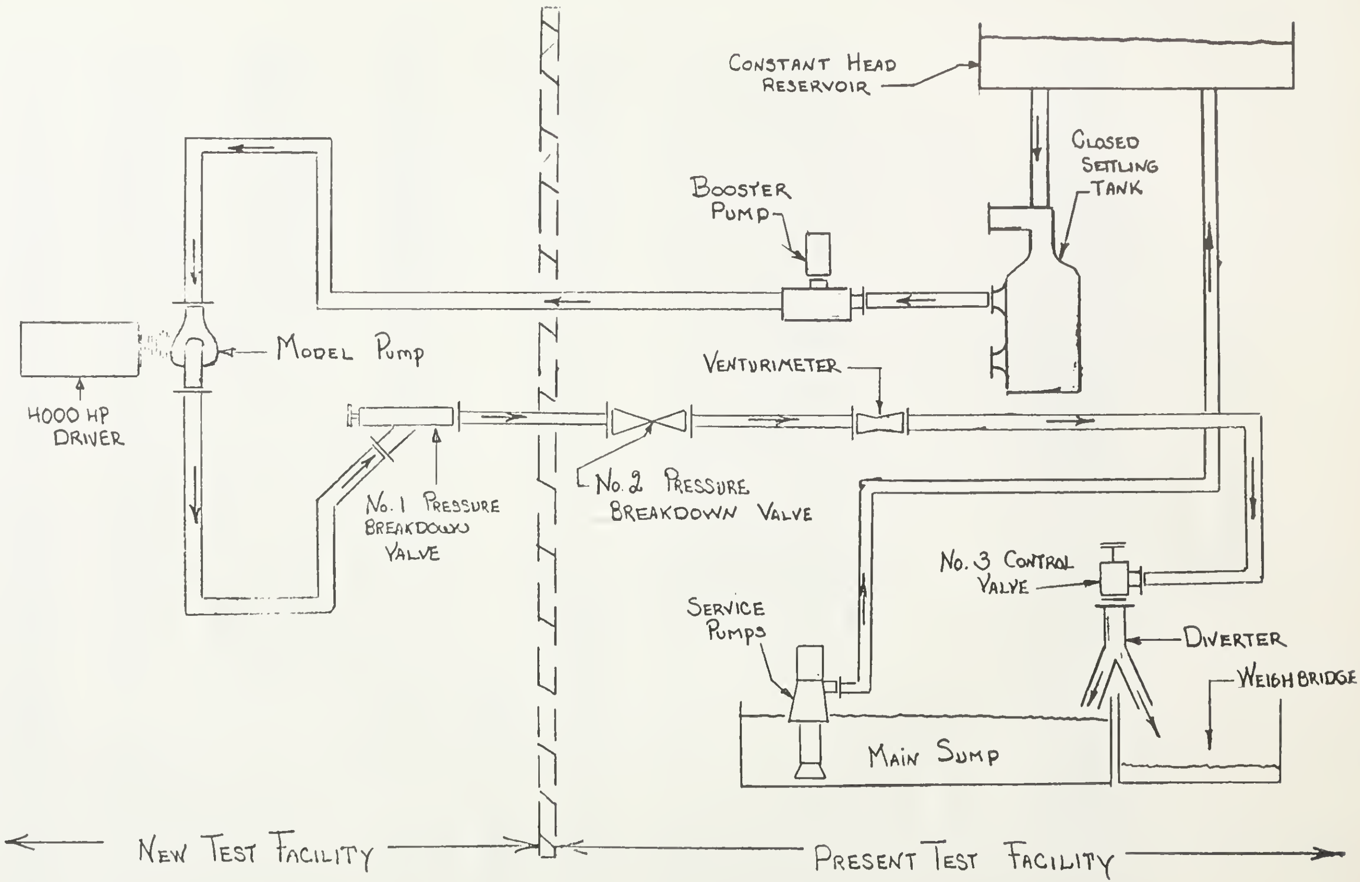

FIG. 2 DIAGRAMMATIC VIEW OF 4000 HP PUMP TEST CIRCUIT

Inter-comparisons between the main and other flow test circuits in NEL have been made to check the reliability of these primary standards. In addition, calibrated flow meters have been exchanged with other Laboratories and any deviations between results have been investigated thoroughly. To ensure that high accuracy would hold for the present tests a minimum diversion time of 30 seconds and a minimum weight of 10,000 lb. (5 tons) was adopted. With a timing repeatability of ± 0.010 second and weighings read to 10 lb this means that the random variations are statistically not expected to exceed ± 0.15 per cent. In fact, as discussed later, they are below ± 0.1 per cent. The systematic bias possible in weigh-bridge calibration, density and timing is also estimated to be less than ± 0.1 per cent and it is concluded that the overall accuracy of flow measurement at NEL can be estimated to be better than ± 0.15 per cent.

In the case of the venturi meters, mercury and water manometers are used to determine the pressure differential. The venturi meters had been calibrated shortly before the tests and thus could be reliably used on their own for cavitation tests on the closed-circuit loop as well as acting as a control on the open-circuit tests. In fact these open-circuit measurements mean that continuous recalibrations are being made on the venturi meters. The discharge coefficients of the two venturi meters, one with an upstream pipe diameter of 18 inches and the other 8 inches, are shown in Fig. 9, expressed in terms of the variation of the discharge coefficient C_d with the Reynolds number Re_d.

$$Q = C_d \frac{A_2}{\sqrt{1 - m^2}} \sqrt{2gh} \text{ cu sec.}$$

$$Re_d = \frac{Q}{\frac{\pi}{4} d \nu}$$

Power Measurements

Torque

The majority of the reactionary torque of the dynamometer carcase was balanced directly by weights on the lever arm. The remainder, the increment between two standard weight values, was supplied by a NEL hydraulic servo capsule acting as a hydraulic vernier. The latter was calibrated over the full torque range immediately prior to each pump test. This calibration is linear in that the reading of capsule pressure is directly proportional to the incremental weight, Fig. 8.

The total torque is therefore given by

$$T = L(W + \frac{R - C}{M})$$

Speed

Whilst an accurate indication of speed is obtained from the speed control gear-box setting this is supplemented by a direct record obtained from an electronic counter. A phonic wheel on the shaft of the dynamometer produces 60 electrical pulses per revolution so that a count taken over a period of 10 seconds provides a figure of 'ten times the revolutions per minute'.

The 10-second period is checked prior to each test series or at any time it is required by direct comparison with the Droitwich transmission of 100 kc/s. The variation has never been more than 6 μs.

Flow Measurement

The Laboratory's primary standard for flow measurement is based on switching the flow for a measured time interval into a 30-ton weigh tank [1]. This method was used for all test-point measurements on open circuit but readings were also taken on manometers linked to the NEL venturimeters [3] installed in the circuit.

The weigh tank is checked by complete loading with dead weights at approximately yearly intervals and by the water-weight method [4] (using a 1-ton load) prior to any important series of tests.

The timing standard is checked against radio frequency signals from the BBC station at Droitwich.

DISCUSSION
of
ALTERNATIVE SCHEMES
for the
TEHACHAPI CROSSING
of the
CALIFORNIA STATE WATER PROJECT

BECHTEL CORPORATION
San Francisco
May 5, 1965

CALIFORNIA STATE WATER PROJECT

TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
DISCUSSION

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CHAPTER I

INTRODUCTION

In January 1965, Bechtel issued a "Second Interim Report on Alternative Schemes for the Tehachapi Crossing of the California State Water Project." This report was based upon a design capacity of 5,000 cfs to provide an annual average flow of 4,650 cfs and upon a range of power costs: a "maximum" of $27.50 per kw capacity plus 3.5 mills per kwh energy; a "minimum" of $20.00 per kw capacity plus 3.0 mills per kwh energy.

After completion of this report, it was learned that DWR was studying three two-lift arrangements which Bechtel had not included in its comparisons. It was considered appropriate that schemes basically similar to these DWR schemes should be so included. Accordingly, upon the same bases as the "Second Interim Report," Bechtel analyzed:

Scheme X - A system utilizing two tunnels and two siphons to by-pass the tunnel and long reservoir of Bechtel's two-lift Pastoria Creek arrangement.

Scheme XI - A system utilizing the ridge route forebay, Bechtel's two-lift intermediate reservoir, then the tunnel-siphon by-pass of Scheme X.

Scheme XII - A two-unequal-lift system with intermediate reservoir on the ridge route.

Pumps were single-stage, single-suction for all three schemes.

This analysis indicated that all three of these schemes were in the same cost range as the schemes included in the "Second Interim Report."

Also subsequent to completion of the January 1965 report: system capacity was reduced to 4,100 cfs to provide an average annual flow of 3,590 cfs; continuous power rates of $16.00 per kw capacity plus 2.0 mills per kwh energy were considered to be available; off-peak power rates of 3.0 mills per kwh energy were considered to be available during the system capacity build-up period; and additional geological studies, particularly in the Pastoria Creek area, were initiated.

As geological studies progressed, conditions were disclosed at the intermediate reservoir of the Pastoria Creek two-lift system that were worse than previously estimated and which would increase construction costs of this system to approach those of the less expensive ridge route schemes. Previous estimates of three-lift schemes, both on Pastoria Creek and on the ridge route already were higher in cost than the less expensive two-lift ridge route schemes. It was further recognized that schemes using balancing tanks are not as desirable from an operating standpoint as those utilizing reservoirs. And, finally, in its report dated April 8, 1965, the Tehachapi Crossing Seismic Board stated:

> "Thus, we find no reason to modify the conclusion expressed in our report of December 22; that is, while the crossing can be effected by either scheme the Ridge scheme is preferable to the Canyon scheme in that it is less vulnerable to damage and presents less potential hazard to life and property."

Bechtel geologists, with some reservations, concur in general with this statement. For these reasons, the studies being presented at this time are confined to three schemes, all on the ridge route:

Scheme I-A	Single-lift, surface discharge pipes, 4-stage, single-suction pumps.
Scheme XII-A	Two-unequal lift, surface discharge pipes, intermediate reservoir, single-stage, single-suction pumps.

Scheme XII-B Same as XII-A except two-stage, double-suction pumps.

CHAPTER II

SCHEMES CONSIDERED

SCHEME I-A

This arrangement is similar to Scheme VIII-A of Bechtel's "Second Interim Report" except that discharge pipes are surface-type. It is also similar to the "DWR single-lift ridge route," as described in the DWR December 1963 report, but is stepped up for the greater capacity. It starts at a forebay about 5,000 feet easterly from Pastoria Creek. For the purposes of this study, a pumping plant containing 14 units (293 cfs, 1991' head, 73,700 HP) was assumed to be located adjacent to the forebay. From this pumping plant, two discharge pipes rise, in open cut, generally southeasterly along a ridge to a combination gate house and surge tank at the portal of Tunnel No. 1 (DWR designation). An alternation of three tunnels and three siphons connects this surge tank to the inlet portal of Tunnel No. 4 of the Tehachapi Crossing.

SCHEME XII-A

This arrangement is a two-unequal-lift scheme, with intermediate reservoir, along the ridge route. It starts at the same forebay as does Scheme I-A. The pumping plant at the forebay contains 8 units (512 cfs, 1106' head, 69,100 HP).

From the pumping plant, two discharge pipes rise along the ridge to a combination gate house and surge tank at a tunnel portal. This tunnel runs easterly to a small reservoir at Elevation 2323 built on a small creek. Debris dam protection is provided for the reservoir. At the reservoir, a second pumping plant, containing 8 units (512 cfs, 882' head, 55,100 HP), is located. From this pumping plant, two discharge pipes rise up to the ridge, then along the ridge, to the same combination gate house and surge tank, and the same sequence of three tunnels and three siphons, as is used for all ridge route schemes. This arrangement is shown in Plan and Profile on Plate 2.

SCHEME XII-B

This scheme is identical with Scheme XII-A except that pumps in both pumping plants are two-stage, double-suction set horizontally instead of the single-stage, single-suction vertical pumps used for Scheme XII-A. This makes the two pumping plants, sucessively: 512 cfs, 1107' head, 70,700 HP; 512 cfs, 883' head, 56,400 HP.

A tabulation of the "Principal Engineering Data" for these three schemes is presented on the following page.

PRINCIPAL ENGINEERING DATA

Scheme Number	I-A	XII-A	XII-B
Number of Lifts	1	2	2
Route	Ridge	Ridge	Ridge
Maximum Capacity: cfs	4,100	4,100	4,100
Average Pumping Rate: cfs	3,850	3,850	3,850
Average Annual Demand: cfs	3,590	3,590	3,590
Canal Normal Water Surface Elevation; feet	1,239	1,239	1,239
Forebay			
Maximum Water Surface Elevation: feet	1,239	1,239	1,239
Normal Water Surface Elevation: feet	1,234	1,234	1,234
Minimum Water Surface Elevation; feet	1,229	1,229	1,229
Area at Normal Water Surface Elevation; acres	35	35	35
Storage for Shutdown; minutes full flow	30	30	30
Storage for Start-up; minutes full flow	30	30	30
Pumping Plant No. 1			
Main Units			
Number of Units	14	8	8
Capacity per Unit; cfs	293	512	512
Rated Head: feet	1,991	1,106	1,107
Number of Pump Stages	4	1	2
Elevation of ℄ of Discharge Pipes: feet	1,159	1,159	1,168
* Minimum Submergence; feet	70	70	61
Balancing Units			
Number of Units	None	4	4
Capacity per Unit; cfs	-	61	61
Pumping Plant No. 2			
Main Units			
Number of Units	None	8	8
Capacity per Unit; cfs	None	512	512
Rated Head; feet	None	882	883
Number of Pump Stages	None	1	2
Elevation of ℄ of Discharge Pipes: feet	None	2,239	2,257
* Minimum Submergence: feet	None	79	61
Balancing Units			
Number of Units	None	None	None
Capacity per Unit; cfs	None	None	None
Discharge Pipes for Pumping Plant No. 1			
Type	Surface	Surface	Surface
Number of Pipes	2	2	2
Maximum Pipe Diameter; feet	14.5	14.5	14.5
Minimum Pipe Diameter; feet	11.5	12.5	12.5
Maximum Steel Thickness; inches	3.14	1.91	1.91
Minimum Steel Thickness; inches	0.64	0.63	0.63
Discharge Pipes for Pumping Plant No. 2			
Type	None	Surface	Surface
Number of Pipes	None	2	2
Maximum Pipe Diameter; feet	None	14.5	14.5
Minimum Pipe Diameter; feet	None	12.5	12.5
Maximum Steel Thickness: inches	None	1.54	1.54
Minimum Steel Thickness: inches	None	0.61	0.61
Number of Combination Surge Tank and Gate House	1	2	2
Tunnels			
Number at 20.0 feet Diameter	3	4	4
Total Length at 20.0 feet Diameter; feet	16,310	17,980	17,980
Siphons			
Number	3	3	3
Total Length	3,620	3,620	3,620
Reservoir No. 1			
Average Water Surface Elevation; feet	None	2,323	2,323
Minimum Operating Water Surface Elevation; feet	None	2,318	2,318
Spillway Crest Elevation; feet	None	2,328	2,328
Maximum Flood Elevation; feet	None	2,338	2,338
Maximum Height of Dam; feet	None	200	200
Length of Dam; feet	None	650	650
Type of Dam	None	Rockfill	Rockfill
Dam Upstream Slope; H:V	None	2.5:1	2.5:1
Dam Downstream Slope; H:V	None	2.25:1	2.25:1
Spillway Type	None	Chute	Chute
Spillway Capacity; cfs	None	4,100	4,100
Debris Dam No. 1			
Storage Capacity; Cu.Yd.	None	10,000	10,000
Energy Gradient at End of System; feet	3,150	3,150	3,150
Total Static Lift; feet	1,886	1,886	1,886
Total Hydraulic Losses; feet	105	102	104

* Measured in distance above ℄ of discharge pipe

CHAPTER III

GEOLOGY

After Bechtel's Second Interim Report was issued in January 1965, DWR started an intensive geological study of the Pastoria Creek area. This study included trenching, test pits and drill holes. Bechtel geologists reviewed the results of these additional explorations and interpreted these results independently. Bechtel also made some additional geological studies of its own, including some drill holes to supplement the DWR program in areas where this was considered advisable. The results of these investigations are being presented in a separate geological report.

CHAPTER IV

BASES OF COMPARATIVE DESIGN

Forebay

Criteria for the design of the forebay remain unchanged from those of the "Second Interim Report" except as affected by the design capacity change from 5,000 cfs to 4,100 cfs. This results in an area change from 42 acres to 35 acres, an inlet crest length reduction from 1000 feet to 800 feet and a spillway crest length reduction from 400 feet to 330 feet. Accordingly, the forebay used herein has 30 minutes of full-flow storage in a 5-foot depth change; either up, for flow rejection, or down, for flow start-up. Embankment heights and spillway provisions are capable of handling even the extremely remote possibility of failure of closure of all discharge valves, simultaneously with complete power failure.

As before, a basic point of comparison 350 feet westerly from Pastoria Creek was selected, and the cost of any main canal easterly from this comparison point is included in the estimates.

Pumping Plants

Pumps in the single-lift system analyzed herein are assumed to be four-stage, single-suction vertical centrifugal pumps without alternatives. Pumps in the two-lift systems are assumed to be single-stage, single-suction vertical pumps for Scheme XII-A, with the alternative of two-stage, double-suction horizontal pumps for Scheme XII-B. The number of units was tentatively selected as fourteen (293 cfs each) for the single-lift system and eight (512 cfs each) for the two-lift systems. This keeps unit capacities reasonably close to (7 or 8% less than) those used for systems in the "Second Interim Report." Future study may indicate that some other number of units is to be preferred, but cost comparisons will not be affected by such a change.

Pumping plant layouts are still not refined sufficiently for final design, but have been given some additional study. As a result, a plant having two parallel rows of units has been adopted as being appreciably less expensive than the more customary single row of units. The layout of such a plant is shown on Plate No. 3, in its application to a two-lift forebay location. It will be noted that this layout incorporates small level-balancing pumps for reasons detailed in the "Second Interim Report." Sizing for these is tentative, but is believed to be such that, should it be so desired, these small units can be installed as pump-turbines and thereby be used to start the larger units by the synchronous starting method.

Discharge Pipes

All discharge pipes considered herein are of the surface type. Two pipes were tentatively selected to be connected to each pumping plant, thus connecting seven units (single-lift system) or four units (two-lift systems) to each pipe. Under the revised flow quantity and power cost criteria, pipe diameters and metal thicknesses remain within feasible limits.

Pipe diameters were determined by the economic study method outlined in the "Second Interim Report." The revised power rates were $16.00 per kw capacity plus 2.0 mills per kwh of energy for continuous power, 3.0 mills per kwh of energy for off-peak power. Off-peak power was used during the system capacity build-up period only, and then for a maximum flow of only 2,050 cfs. Base date was revised to January 1, 1965. Pumping plant construction costs and operating costs were considered, as were also differences in discharge pipe costs. As a result, the value per foot of conveyance losses was redetermined to be $212,000.

Pipe manifolds at pumping plants were similarly analyzed for economic sizes, and the costs so determined were included in discharge pipe costs.

The same steels were used as were used for the "Second Interim Report."

Surge Tanks

The combination gate house and surge tank arrangements used in the "Second Interim Report" was used herein without cost change.

Tunnels

Economic analysis of tunnels was revised to suit the conveyance head loss value of $212,000 per foot. As a result all tunnels are 20.0 feet, inside diameter.

Siphons

Ridge route siphon designs were not reviewed at this time and costs used in the "Second Interim Report" were stepped down in proportion to system capacity.

Dam

Dam design is of a preliminary nature based upon the most up-to-date geological investigations. It is a center-core, rock-fill dam, basically sloped 2.5:1 upstream and 2.25:1 downstream. Spillway is a free-crested open-channel chute. Reservoir capacity, between "normal" and "critical" submergence points of the applicable pumps, is about 30 minutes of 4,100 cfs flow.

Debris Dam

Drainage area is so small that debris dam cost is considered to be nominal.

CHAPTER V

COMPARATIVE CONSTRUCTION COSTS

The estimated comparative construction costs of the alternative schemes considered herein are summarized as follows:

Scheme No.	No. of Lifts	Pump Type	Comparative Construction Cost in Millions of Dollars: Total	Difference
I-A	1	4S, SS-V	108.8	+ 2.1
XII-A	2	1S, SS-V	106.7	Base
XII-B	2	2S, DS-H	120.0	+13.3

All schemes are on the ridge route and use surface discharge pipelines.

As for the "Second Interim Report," the above total estimated comparative construction costs include an allowance of 30% for engineering and contingencies. However, no allowance is included for land costs, for rights-of-way, for escalation or for interest during construction, so these costs are incomplete and are comparable only with each other. Moreover, these costs are based upon 1965 prices with construction "all at once" for the same reasons as were used for the "Second Interim Report."

The comparative construction costs of the three schemes are presented in more detail on the following pages.

COMPARATIVE CONSTRUCTION COSTS SUMMARIZED BY PRINCIPAL FEATURES

Scheme No. No. of Lifts Route	Costs in Millions of Dollars I-A 1 Ridge	XII-A 2 Ridge	XII-B 2 Ridge
Pumping Plants	39.3	38.3	47.9
Discharge Pipes	18.3	11.5	12.1
Tunnels	16.3	18.0	18.0
Siphons	5.9	5.9	5.9
Dams	None	2.9	2.9
Forebays, Surge Tanks	3.9	4.7	4.7
Access Roads	None	0.8	0.8
Subtotal	83.7	82.1	92.3
Eng. & Cont. @ 30%	25.1	24.6	27.7
Total Construction Cost	108.8	106.7	120.0

SUMMARY OF ESTIMATED CONSTRUCTION COSTS
IN THOUSANDS OF DOLLARS

Scheme Number	I-A	XII-A	XII-B
Number of Lifts	1	2	2
Route Location	Ridge	Ridge	Ridge
Pump Type	4S,SS-V	1S,SS-V	2S,DS-H
Forebay	3,053	3,053	3,053
Pumping Plant No. 1	39,301	20,746	25,776
Discharge Pipes	18,301	7,716	8,137
Surge Tank No. 1	800	800	800
Tunnel No. 1	7,479	7,479	7,479
Siphon No. 1	645	645	645
Dam No. 1	None	2,784	2,784
Debris Dam No. 1	None	100	100
Pumping Plant No. 2	None	17,510	22,186
Discharge Pipes	None	3,797	3,944
Surge Tank No. 2	None	800	800
Tunnel No. 2	2,744	2,744	2,744
Siphon No. 2	3,328	3,328	3,328
Tunnel No. 3	6,045	6,045	6,045
Siphon No. 3	1,973	1,973	1,973
Tunnel No. 1-A	None	1,712	1,712
Access Roads	None	818	818
Sub-Total	83,669	82,050	92,324
Eng. & Cont. @ 30%	25,101	24,615	27,697
TOTAL	108,770	106,665	120,021

CHAPTER VI

COMPARATIVE ANNUAL OPERATING COSTS

The estimated comparative annual operating costs, including differences in conveyance losses as well as differences in the efficiencies of different types of pumps, of the alternative schemes considered herein are summarized below:

Scheme No.	No. of Lifts	Pump Type	Comparative Annual Operating Cost in Millions of Dollars: Total	Difference
I-A	1	4S, SS-V	27.9	+1.1
XII-A	2	1S, SS-V	26.8	Base
XII-B	2	2S, DS-H	27.5	+0.7

All schemes are on the ridge route and use surface discharge pipelines.

The above annual operating costs are based upon the following assumed equipment efficiencies:

Item	Efficiency in %
Transformers, all systems	98.25*
Motors, one-lift systems	97.8
Motors, two-lift systems	97.7
Pumps, 1S, SS	93.0
Pumps, 2S, DS	91.0
Pumps 4S, SS	90.0

* Actual efficiencies are reduced 1.0% to allow for auxiliaries.

These equipment efficiencies result in the following overall system pumping efficiencies:

Scheme No.	No. of Lifts	Pump Type	Efficiency %
I-A	1	4S, SS-V	86.5
XII-A	2	1S, SS-V	89.3
XII-B	2	2S, DS-H	87.4

Continuous power cost is based upon a capacity rate of $16.00 per kw per year plus an energy rate of 2.0 mills per kwh. Off-peak power cost is based upon no capacity charge but an energy rate of 3.0 mills per kwh. Power costs are based upon 8,150 hours of operation per year at an average pumping rate of 3,850 cfs. Annual operating costs also include operating, maintenance and replacement charges of:

$3.40 per kw per year for single-lift pumping plant;

$3.25 per kw per year for two-lift pumping plants using single-stage, single-suction pumps;

$3.45 per kw per year for two-lift pumping plants using two-stage, double-suction pumps;

0.75% of construction cost per year for surface discharge pipes;

0.25% of construction cost per year for all other structures.

All costs are based upon complete installations operating at the full average pumping rate. Off-peak power is used only during the build-up period, and then only to a maximum pumping rate of 2,050 cfs, and therefore does not enter into the computations of results shown in this table.

Pumping plant operating and maintenance costs are based upon FPC recommendations for similar-sized hydroelectric plants operating in Southern California. Replacement costs are based upon MWD experience. These bases are obviously applicable only to pumping plants utilizing single-stage, single-suction pumps. Accordingly, an adjustment for applicability to multi-stage pumps was made by assuming that the basic costs include 5¢ per kw per year for maintenance and 10¢ per kw per year for replacement of single-stage pumps. These two figures were increased by approximate pump cost ratios (2.5 for two-stage, double-suction; 3.0 for four-stage, single-suction pumps) to allow for the additional maintenance and replacement costs for multi-stage pumps.

The estimated conveyance losses included in the estimated comparative annual operating costs in the above tabulation are for the average pumping rate of 3,850 cfs and are as follows:

Scheme No.	No. of Lifts	Pump Type	Estimated Conveyance Loss-Feet Total	Difference
I-A	1	4S, SS-V	90	+3
XII-A	2	1S, SS-V	87	Base
XII-B	2	2S, DS-H	89	+2

All schemes are on the ridge route and use surface discharge pipelines.

It should be noted that differences in conveyance losses are small, especially when compared with the total pumping head of some 1,980 feet.

These comparative annual operating costs are presented in more detail on the following page.

SUMMARY OF ESTIMATED ANNUAL OPERATING COSTS
IN THOUSANDS OF DOLLARS

Alternative Number	I-A	XII-A	XII-B
Number of Lifts	1	2	2
Route Location	Ridge	Ridge	Ridge
Pump Type	4S, SS-V	1S, SS-V	2S, DS-H
Pumping Plant	1 @ 14 Units	2 @ 8 Units	2 @ 8 Units
System Capacity - cfs	4,100	4,100	4,100
Average Pumping Rate - cfs	3,850	3,850	3,850
System Conveyance Losses - ft. for 4100 cfs	105	102	104
System Conveyance Losses - ft. for 3850 cfs.	90	87	89
System Static Head - ft.	1,886	1,886	1,886
Varying Costs			
Pumping Plant, O&M	2,717	2,512	2,669
Debris Removal	-	2	2
Other Structures, O&M	263	217	223
Conv. Losses - Capacity @ $16.00/KW	674	634	661
Conv. Losses - Energy @ 2.0 Mills/KWH	553	518	541
Total Varying Costs	4,207	3,883	4,096
Fixed Costs			
Static Head - Capacity @ $16.00/KW	12,110	11,730	11,985
Static Head - Energy @ 2.0 Mills/KWH	11,585	11,222	11,466
Total Fixed Costs	23,695	22,952	23,451
Total Operating Costs	27,902	26,835	27,547

Rev.

CHAPTER VII

COMPARATIVE TOTAL COSTS

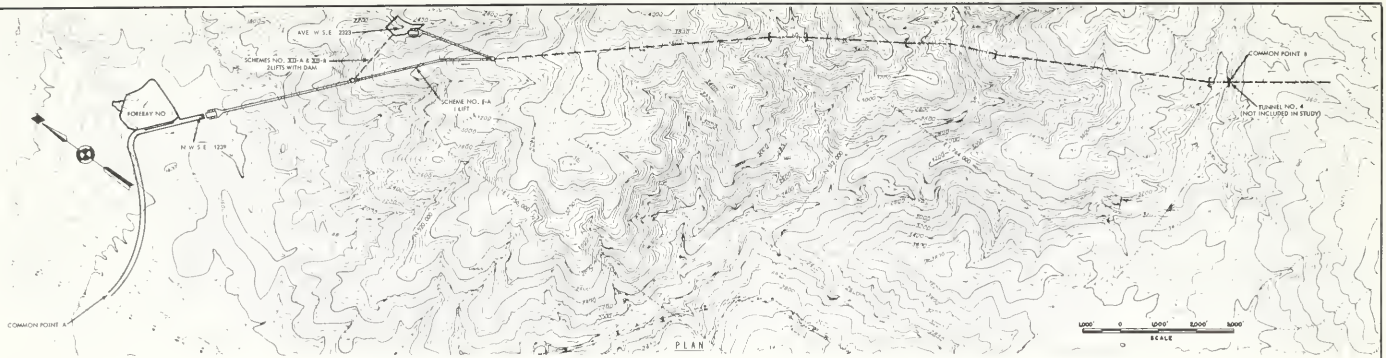
There are many ways to compare the total costs of the various schemes under consideration. The more sophisticated methods require lengthy computations which are not justifiable, nor even more accurate, for preliminary study comparisons of the type under consideration. Accordingly, an abbreviated method has been adopted which is believed to be at least as accurate as the information upon which it is based.

As in the "Second Interim Report," it is considered that construction costs, on an "all at once" basis, validly represent the present value of bond interest and amortization. Therefore, differences in these construction costs represent the present value of differences in bond payments. Consequently, in order to determine the present value of differences in total costs, it is only necessary to determine the present value of operating cost differentials. By making a trial run for a typical case involving a water build-up period utilizing some off-peak power from 1971 to 1991 and full operation from 1991 through 2040, it has been determined that a factor of 13 is a reasonable value for converting annual operating cost differentials to the appropriate present value. Accordingly, the cost differentials shown in Chapters V and VI are here combined, using the factor of 13 applied to annual operating cost differentials, to show the present value of the cost differences between the various schemes.

ESTIMATED COMPARATIVE TOTAL COST DIFFERENTIALS

			Comparative Cost Differentials in Millions of Dollars			
Scheme No.	No.of Lifts	Pump Type	Constr. Cost	Annual Oper. Cost	Present Worth of Oper. Cost	Present Worth of Total Cost
XII-A	2	1S, SS-V	Base	Base	Base	Base
I-A	1	4S, SS-V	+2.1	+1.1	+14.3	+16.4
XII-B	2	2S, DS-H	+13.3	+0.7	+9.1	+22.4

All schemes are on the ridge route and use surface discharge pipelines.



LOCATION MAP

LEGEND

PUMPING PLANT
DISCHARGE PIPES
SURGE TANK
TUNNEL
SIPHON
DAM AND RESERVOIR

NOTES

TOPOGRAPHY OBTAINED FROM MAP COMPILED BY STEREOPHOTOGRAMMETRIC METHODS FROM PHOTOGRAPHY DATED JULY 21, 1956

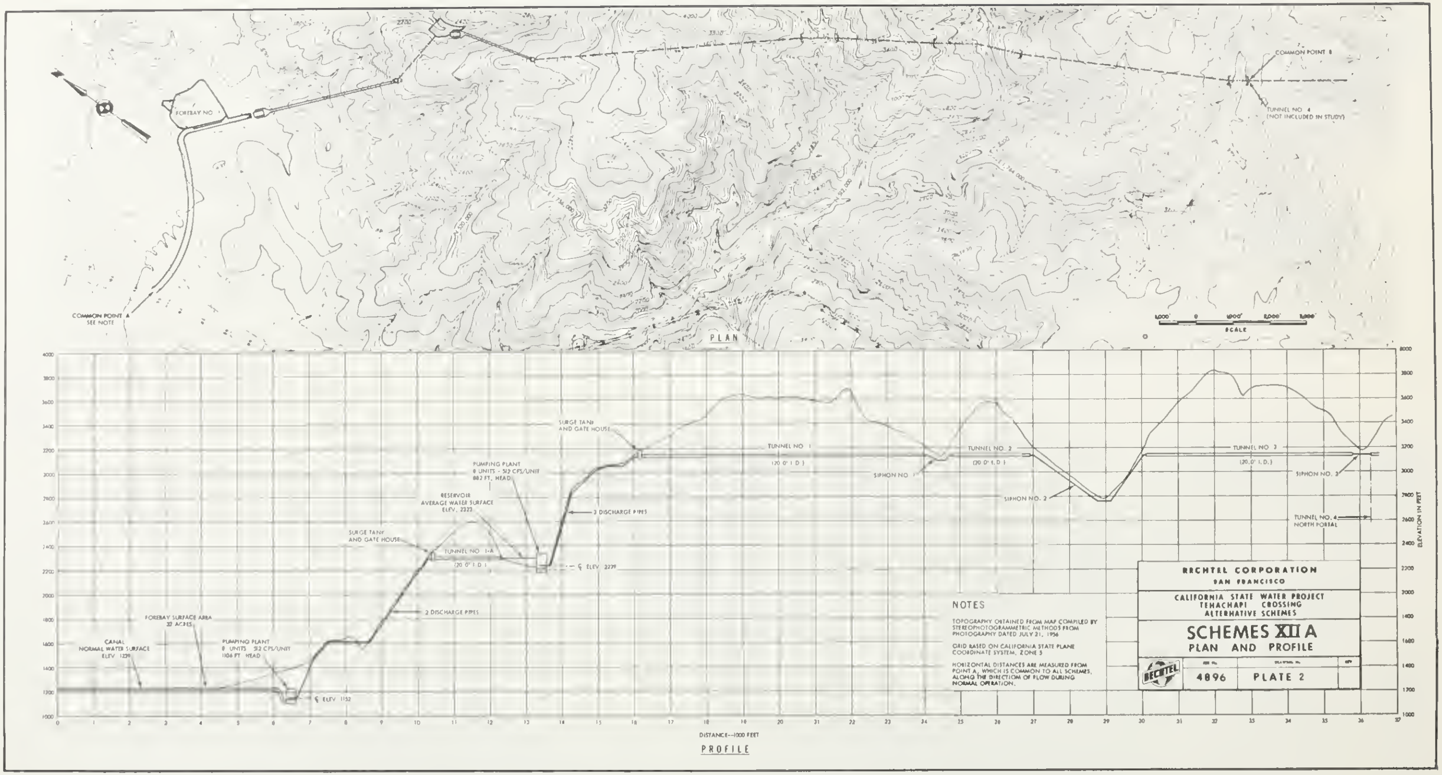
GRID BASED ON CALIFORNIA STATE PLANE COORDINATE SYSTEM, ZONE 5

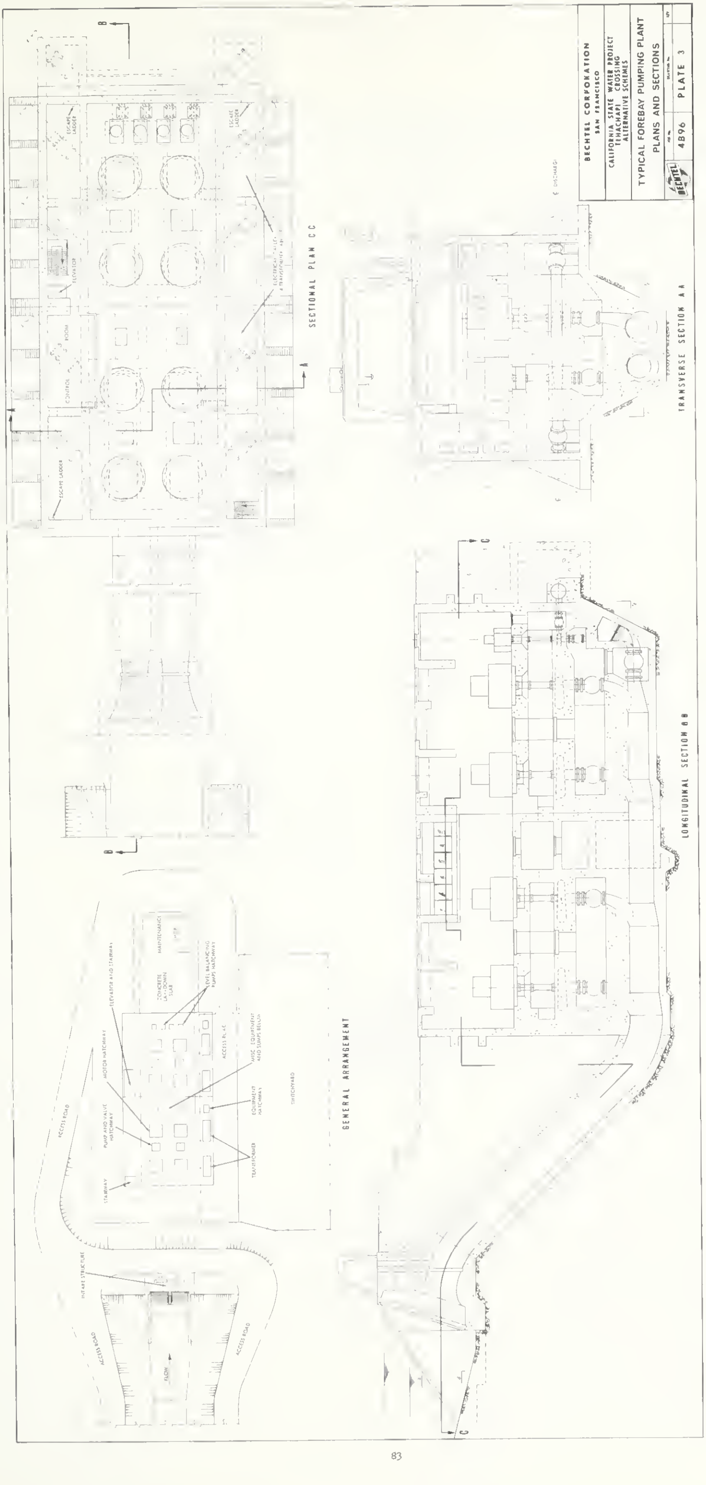
BECHTEL CORPORATION
SAN FRANCISCO

CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES

LOCATION MAP
AND COMPARISON OF PLANS

4896 PLATE 1


COMMON POINT B
TUNNEL NO. 4
(NOT INCLUDED IN STUDY)
FOREBAY NO.
COMMON POINT A
SEE NOTE
SCALE
PLAN
SURGE TANK
AND GATE HOUSE
TUNNEL NO. 1
(20.0' I.D.)
TUNNEL NO. 2
(20.0' I.D.)
TUNNEL NO. 3
(20.0' I.D.)
SIPHON NO. 1
SIPHON NO. 2
SIPHON NO. 3
TUNNEL NO. 4
NORTH PORTAL
PUMPING PLANT
8 UNITS - 512 CFS/UNIT
882 FT. HEAD
RESERVOIR
AVERAGE WATER SURFACE
ELEV. 2322
3 DISCHARGE PIPES
SURGE TANK
AND GATE HOUSE
TUNNEL NO. 1-A
(20.0' I.D.)
℄ ELEV. 2279
2 DISCHARGE PIPES
FOREBAY SURFACE AREA
32 ACRES
CANAL
NORMAL WATER SURFACE
ELEV. 1239
PUMPING PLANT
8 UNITS 512 CFS/UNIT
1106 FT. HEAD
℄ ELEV. 1152
ELEVATION IN FEET
DISTANCE—1000 FEET
PROFILE
NOTES
TOPOGRAPHY OBTAINED FROM MAP COMPILED BY STEREOPHOTOGRAMMETRIC METHODS FROM PHOTOGRAPHY DATED JULY 21, 1956
GRID BASED ON CALIFORNIA STATE PLANE COORDINATE SYSTEM, ZONE 5
HORIZONTAL DISTANCES ARE MEASURED FROM POINT A, WHICH IS COMMON TO ALL SCHEMES, ALONG THE DIRECTION OF FLOW DURING NORMAL OPERATION.
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEMES XII A
PLAN AND PROFILE
4896
PLATE 2


BECHTEL CORPORATION
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CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
TYPICAL FOREBAY PUMPING PLANT
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4B96
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ELEVATOR
CONTROL ROOM
ACCESS ROAD
INTAKE STRUCTURE
FLOW
STAIRWAY
PUMP AND VALVE HATCHWAY
MOTOR HATCHWAY
ELEVATOR AND STAIRWAY
EQUIPMENT HATCHWAY
TRANSFORMER
SWITCHYARD

SUPPLEMENT

to

SECOND INTERIM REPORT

on

ALTERNATIVE SCHEMES

for the

TEHACHAPI CROSSING

of the

CALIFORNIA STATE WATER PROJECT

for

THE METROPOLITAN WATER DISTRICT OF SOUTHERN CALIFORNIA

Prepared By

BECHTEL CORPORATION

SAN FRANCISCO

APRIL 1965

CALIFORNIA STATE WATER PROJECT

TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SUPPLEMENT TO SECOND INTERIM REPORT

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CHAPTER I

INTRODUCTION

After the Bechtel report "Second Interim Report on Alternative Schemes for the Tehachapi Crossing of the California State Water Project" was completed, it was learned that DWR is studying three two-lift arrangements which Bechtel had not included in its comparisons. It was considered appropriate that similar schemes should be so included. Accordingly, Bechtel has now studied a system utilizing tunnels and a siphon instead of the long reservoir included in Bechtel's two-lift Pastoria Creek arrangement; a system starting at the ridge route forebay, crossing to Bechtel's two-lift intermediate reservoir, then by-passing the long reservoir via tunnels and a siphon; and a two-unequal-lift system with intermediate reservoir on the ridge route.

These systems were analyzed in the same manner and on identical bases as were the schemes for the Second Interim Report so that the results are comparable in every detail. Design flow remains at 5,000 cfs. Pumps are single-stage, single-suction for all three schemes. Power costs are the same as those used in the Second Interim Report.

CHAPTER II

SCHEMES CONSIDERED

This supplement is concerned only with the three additional arrangements (Schemes X, XI and XII). No variations of any of these schemes have been considered.

SCHEME X

This arrangement is essentially a modification of Bechtel Scheme III-A, a two-equal-lift scheme along Pastoria Creek. It starts at the same forebay, near the mouth of Pastoria Creek. Both pumping plants contain 9 units each (556 cfs, 980 ft. head, 66,500 HP). From the pumping plant at the forebay, three discharge pipes rise along a ridge to a combination gate house and surge tank at a tunnel portal. This tunnel runs generally southeasterly to discharge into a reservoir at Elevation 2192 on Pastoria Creek slightly more than halfway to the entrance of DWR Tunnel No. 4. From this reservoir, the second pumping plant discharges through three discharge pipes up the southerly canyon wall through a combination gate house and surge tank into a tunnel portal. This tunnel runs southeasterly, changes to a siphon to cross under Pastoria Creek, then continues as a tunnel to the beginning of DWR Siphon No. 3 and thence to connect to the entrance of Tunnel No. 4. This arrangement is shown in Plan and Profile on Plate 3.

SCHEME XI

This arrangement is a two-equal-lift scheme combining both ridge route and Pastoria Creek in its siting. It starts at the ridge route forebay (Forebay No. 1). Both pumping plants contain 9 units each (556 cfs, 979 ft. head, 66,500 HP). From the pumping plant at the forebay, three discharge pipes rise along the ridge route to a combination gate house and surge tank at a tunnel portal. This tunnel runs generally southerly to discharge into the same reservoir used for Scheme X, except that reservoir level is at Elevation 2193. From this reservoir, the arrangement is identical with Scheme X. This arrangement is shown in Plan and Profile on Plate 4.

SCHEME XII

This arrangement is a two-unequal-lift scheme, with intermediate reservoir, along the ridge route. It starts at the same forebay as does Scheme XI. The pumping plant at the forebay contains 9 units (556 cfs, 1099 ft head, 74,600 HP). From the pumping plant, three discharge pipes rise along the ridge to a combination gate house and surge tank at a tunnel portal. This tunnel runs easterly to a small reservoir at Elevation 2320 built on a small creek. Debris dam protection is provided for the reservoir. At the reservoir, a second pumping plant, containing 9 units (556 cfs, 861 ft. head, 58,500 HP), is located. From this pumping plant, three discharge pipes rise up to the ridge, then along the ridge, to the same combination gate house and surge tank, and the same sequence of three tunnels and three siphons, as is used for all ridge route schemes. This arrangement is shown in Plan and Profile on Plate 5.

A tabulation of the "Principal Engineering Data" for these three schemes is presented on the following page.

Scheme Number	X	XI	XII
Number of Lifts	2	2	2
Route	Pastoria	Pastoria & Ridge	Ridge
Maximum Capacity; cfs	5,000	5,000	5,000
Average Demand; cfs	4,650	4,650	4,650
Canal Normal Water Surface Elevation; feet	1,239	1,239	1,239
Forebay			
Maximum Water Surface Elevation; feet	1,239	1,239	1,239
Normal Water Surface Elevation; feet	1,234	1,234	1,234
Minimum Water Surface Elevation; feet	1,229	1,229	1,229
Area at Normal Water Surface Elevation; acres	42	42	42
Storage for Shutdown; minutes full flow	30	30	30
Storage for Start-up; minutes full flow	30	30	30
Pumping Plant No. 1			
Main Units			
Number of Units	9	9	9
Capacity per Unit; cfs	556	556	556
Rated Head; feet	980	979	1,099
Number of Pump Stages	1	1	1
Elevation of ℄ of Discharge Pipes; feet	1,149	1,149	1,151
* Minimum Submergence; feet	80	80	78
Balancing Units			
Number of Units	4	4	4
Capacity per Unit; cfs	70	70	70
Pumping Plant No. 2			
Main Units			
Number of Units	9	9	9
Capacity per Unit; cfs	556	556	556
Rated Head; feet	980	979	861
Number of Pump Stages	1	1	1
Elevation of ℄ of Discharge Pipes; feet	2,110	2,111	2,231
* Minimum Submergence; feet	80	80	87
* Critical Submergence; feet	--	--	--
Balancing Units			
Number of Units	None	None	None
Capacity per Unit; cfs	None	None	None
Discharge Pipes for Pumping Plant No. 1			
Type	Surface	Surface	Surface
Number of Pipes	3	3	3
Maximum Pipe Diameter; feet	14.5	14.5	14.5
Minimum Pipe Diameter; feet	12.0	12.0	12.0
Maximum Steel Thickness; inches	1.62	1.62	1.77
Minimum Steel Thickness; inches	0.61	0.61	0.61
Discharge Pipes for Pumping Plant No. 2			
Type	Surface	Surface	Surface
Number of Pipes	3	3	3
Maximum Pipe Diameter; feet	14.5	14.5	14.5
Minimum Pipe Diameter; feet	12.0	12.0	12.0
Maximum Steel Thickness; inches	1.62	1.62	1.47
Minimum Steel Thickness; inches	0.61	0.61	0.61
Number of Combination Surge Tank and Gate House	2	2	2
Tunnels			
Number at 24.5 feet Diameter	3	3	3
Total Length at 24.5 feet Diameter; feet	19,920	20,980	12,550
Number at 24.0 feet Diameter	None	None	1
Total Length at 24.0 feet Diameter	--	--	5,570
Siphons			
Number	2	2	3
Total Length	2,040	2,040	3,620
Reservoir No. 1			
Normal Water Surface Elevation; feet	2,192	2,193	2,320
Minimum Operating Water Surface Elevation; feet	2,190	2,191	2,317
Spillway Crest Elevation; feet	2,200	2,201	2,327
Maximum Flood Elevation; feet	2,215	2,216	2,337
Maximum Height of Dam; feet	140	140	170
Length of Dam; feet	605	605	600
Type of Dam	Rockfill	Rockfill	Rockfill
Dam Upstream Slope; H:V	2.5:1	2.5:1	2.5:1
Dam Downstream Slope; H:V	2.25:1	2.25:1	2.25:1
Spillway Type	Morning Glory	Morning Glory	Chute
Spillway Capacity; cfs	39,000	39,000	5,000
Debris Dam No. 1			
Storage Capacity; Cu.Yd.	30,400	30,400	None
Hydraulic Gradient at End of System; feet	3,150	3,150	3,150
Total Static Lift; feet	1,911	1,911	1,911
Total Hydraulic Losses; feet	49	47	49

* Measured in distance above ℄ of discharge pipe

CHAPTER III

GEOLOGY

After Bechtel's Second Interim Report was issued in January 1965, DWR started an intensive geological study of the Pastoria Creek area. This study includes trenching, test pits and drill holes. Bechtel geologists are reviewing the results of these additional explorations and are interpreting these results independently. Bechtel also is making some additional geological studies of its own, including some drill holes to supplement the DWR program in areas where this was considered advisable. These investigations are still in progress and the results are not yet available in final form. They will be presented in a separate geological report in the near future.

The studies of alternative schemes in this supplement to the Second Interim Report are based upon the geological information available in January 1965, and do not incorporate the results of the additional geological investigations.

CHAPTER IV

BASES OF COMPARATIVE DESIGN

For this supplement, the bases of comparative design are identical with those used for the "Second Interim Report". Plant capacity remains at 5,000 cfs. Single-stage, single-suction pumps were used for all three schemes. Level balancing pumps were used in the forebay pumping plant of each system. Value of one foot of head remains at $350,000. Design factors, and other comparative factors, are unchanged.

CHAPTER V

COMPARATIVE CONSTRUCTION COSTS

The estimated comparative construction costs of the alternative schemes considered herein are summarized in comparison with those considered in the "Second Interim Report" as follows:

Scheme No.	Route & No. of Lifts	Pump Type	Comparative Construction Cost In Millions of Dollars Total	Difference
II-A	P-3-S	1S, SS-V	140.3	+12.5
III-A	P-2-S	1S, SS-V	127.8	Base
V-A	R-2-S	1S, SS-V	133.4	+ 5.6
V-B	R-2-S	2S, DS-V	141.1	+13.3
V-C	R-2-S	2S, DS-H	140.8	+13.0
VIII-A	R-1-US	4S, SS-V	139.6	+11.8
IX	R-3-S	1S, SS-V	148.3	+20.5
X	P-2-S	1S, SS-V	131.2	+ 3.4
XI	RP-2-S	1S, SS-V	132.8	+ 5.0
XII	R-2-S	1S, SS-V	139.2	+11.4

P indicates Pastoria Creek route

R indicates ridge route

RP indicates a combination of ridge and Pastoria Creek routes

S indicates surface discharge pipelines

US indicates a combination of underground and surface discharge pipelines.

As for the "Second Interim Report", the above total estimated comparative construction costs include an allowance of 30% for engineering and contingencies. However, no allowance is included for land costs, for rights-of-way, for escalation or for interest during construction, so these costs are incomplete and are comparable only with each other. Moreover, these costs are based upon 1965 prices with construction "all at once" for the same reasons as were used for the "Second Interim Report".

The comparative construction costs of the three supplementary schemes are presented in more detail on the following pages.

COMPARATIVE CONSTRUCTION COSTS SUMMARIZED BY PRINCIPLE FEATURES

COSTS IN MILLIONS OF DOLLARS

Scheme No. No. of Lifts Route	X 2 Pastoria	XI 2 Ridge & Past.	XII 2 Ridge
Pumping Plants	43.5	44.9	49.4
Discharge Pipes	18.2	15.9	17.4
Tunnels	25.9	27.3	24.1
Siphons	4.8	4.8	7.2
Dams	3.1	3.1	3.1
Balancing Tanks	None	None	None
Forebays, Surge Tanks	4.3	5.1	5.1
Access Roads	1.1	1.1	0.8
Subtotal	100.9	102.2	107.1
Eng. & Cont.@ 30%	30.3	30.6	32.1
TOTAL CONSTRUCTION COST	131.2	132.8	139.2

SUMMARY OF ESTIMATED CONSTRUCTION COSTS
IN THOUSANDS OF DOLLARS

Scheme Number Number of Lifts Route Location	X 2 Past.Creek	XI 2 Ridge & Past.Creek	XII 2 Ridge
Forebay	2,711	3,459	3,459
Pumping Plant No. 1	23,139	24,525	25,494
Discharge Pipes	12,086	9,780	11,206
Surge Tank No. 1	800	800	800
Balancing Tank	None	None	None
Tunnel No. 1	13,767	15,145	10,218
Siphon No. 1	2,374	2,374	786
Dam	2,640	2,640	3,129
Debris Dam	432	432	None
Pumping Plant No. 2	20,409	20,409	23,857
Discharge Pipes	6,145	6,145	6,145
Surge Tank No. 2	800	800	800
Balancing Tank	None	None	None
Tunnel No. 2	4,654	4,654	3,744
Siphon No. 2	2,406	2,406	4,058
Dam	None	None	None
Debris Dam	None	None	None
Pumping Plant No. 3	None	None	None
Discharge Pipes	None	None	None
Surge Tank No. 3	None	None	None
Tunnel No. 3	7,475	7,475	7,798
Siphon No. 3	None	None	2,406
Dam	None	None	None
Debris Dam	None	None	None
Tunnel No. 1-A	None	None	2,353
Tunnel No. 1-B	None	None	None
Access Roads	1,111	1,135	806
Sub-Total	100,949	102,179	107,059
Eng. & Cont. @ 30%	30,285	30,654	32,118
Total	131,234	132,833	139,177

CHAPTER VI

COMPARATIVE ANNUAL OPERATING COSTS

The estimated comparative annual operating costs, including differences in conveyance losses as well as differences in the efficiencies of different types of pumps, of the alternative schemes considered herein are summarized in comparison with those considered in the "Second Interim Report" below:

Scheme No.	Route & No. of Lifts	Pump Type	Total Comparative Annual Operating Cost in Millions of Dollars: Maximum	Minimum	Average Difference
II-A	P-3-S	1S, SS-V	55.6	44.8	+0.25
III-A	P-2-S	1S, SS-V	55.3	44.6	Base
V-A	R-2-S	1S, SS-V	55.5	44.7	+0.15
V-B	R-2-S	2S, DS-V	57.0	45.9	+1.5
V-C	R-2-S	2S, DS-H	56.9	45.9	+1.45
VIII-A	R-1-US	4S, SS-V	57.3	46.2	+1.8
IX	R-3-S	1S, SS-V	55.7	44.9	+0.35
X	P-2-S	1S, SS-V	55.4	44.7	+0.1
XI	RP-2-S	1S, SS-V	55.4	44.6	+0.05
XII	R-2-S	1S, SS-V	55.3	44.6	0

P indicates Pastoria Creek route

R indicates ridge route

RP indicates a combination of ridge and Pastoria Creek routes

S indicates surface discharge pipelines

US indicates a combination of underground and surface discharge pipelines.

The above annual operating costs are based upon the same equipment efficiencies, the same power rates (Maximum $27.50 per kw plus 3.5 mills per kwh; Minimum $20.00 per kw plus 3.0 mills per kwh) and the same operation, maintenance and replacement charges and operating assumptions as were used in the "Second Interim Report".

The estimated conveyance losses included in the above-tabulated costs are, for the additional schemes considered, as follows:

Scheme No.	Estimated Conveyance Loss - Feet Total	Difference*
X	49	+6
XI	47	+4
XII	49	+6

All schemes are two-lift schemes utilizing surface discharge pipelines and single-stage, single-suction, vertical pumps.

*Differences refer to Scheme III-A, the basis used in the Second Interim Report.

The comparative annual operating costs of the three supplementary schemes are presented in more detail on the following page.

SUMMARY OF ESTIMATED ANNUAL OPERATING COSTS
IN THOUSANDS OF DOLLARS

Scheme Number	X	XI	XII
Number of Lifts	2	2	2
Route Location	Pastoria	Ridge & Past.	Ridge
Pumping Plant	2 @ 9 Units	2 @ 9 Units	2 @ 9 Units
System Capacity - cfs	5,000	5,000	5,000
System Conv. Losses - ft.	49	47	49
System Static Head - ft.	1,911	1,911	1,911
Varying Cost			
Pumping Plant, O & M	2,930	2,927	2,930
Debris Removal	85	85	2
Other Structures, O & M	305	289	300
Conv.Losses-Capacity @ $27.50/KW	639	613	639
Conv.Losses-Energy @ 3.5 Mills/KWH	663	636	663
Total "Maximum" Varying Costs	4,622	4,550	4,534
Conv.Losses-Capacity @ $20.00/KW	465	446	465
Conv.Losses-Energy @ 3.0 Mills/KWH	568	545	568
Total "Minimum" Varying Costs	4,353	4,292	4,265
Fixed Costs			
Static Head-Capacity @ $27.50/KW	24,936	24,936	24,936
Static Head-Energy @ 3.5 Mills/KWH	25,866	25,866	25,866
Total "Maximum" Fixed Costs	50,802	50,802	50,802
Static Head-Capacity @ $20.00/KW	18,135	18,135	18,135
Static Head-Energy @ 3.0 Mills/KWH	22,171	22,171	22,171
Total "Minimum" Fixed Costs	40,306	40,306	40,306
Total Operating Costs			
"Maximum"	55,424	55,352	55,336
"Minimum"	44,660	44,598	44,571

CHAPTER VII

COMPARATIVE TOTAL COSTS

The estimated comparative total cost differentials for the schemes considered herein are compared with those in the "Second Interim Report", in the same manner and for the same reasons as used in that report, below:

Scheme No.	Route & No. of Lifts.	Pump Type	Comparative Cost Differentials in Millions of Dollars: Constr. Cost	Annual Operatg. Cost	Present Worth of Operating Cost	Present Worth of Total Cost
III-A	P-2-S	1S, SS-V	Base	Base	Base	Base
X	P-2-S	1S, SS-V	+ 3.4	+ 0.1	+ 1.2	+ 4.6
XI	RP-2-S	1S, SS-V	+ 5.0	+0.05	+ 0.6	+ 5.6
V-A	R-2-S	1S, SS-V	+ 5.6	+0.15	+ 1.8	+ 7.4
XII	R-2-S	1S, SS-V	+11.4	0	0	+11.4
II-A	P-3-S	1S, SS-V	+12.5	+0.25	+ 3.0	+15.5
IX	R-3-S	1S, SS-V	+20.5	+0.35	+ 4.2	+24.7
V-C	R-2-S	2S, DS-H	+13.0	+1.45	+17.4	+30.4
V-B	R-2-S	2S, DS-V	+13.3	+ 1.5	+18.0	+31.3
VIII-A	R-1-US	4S, SS-V	+11.8	+ 1.8	+21.6	+33.4

P indicates Pastoria Creek route

R indicates ridge route

RP indicates a combination of ridge and Pastoria Creek routes

S indicates surface discharge pipelines

US indicates a combination of underground and surface discharge pipelines.

As before, it should be ncted that all arrangements utilizing multi-stage pumps are more expensive in total cost than are the schemes utilizing single-stage pumps: also that ridge route schemes are more expensive, type for type, than Pastoria Creek schemes.

CHAPTER VIII

DISCUSSION

Scheme X is a two-lift Pastoria Creek arrangement differing from Scheme III-A only in that two tunnels and two siphons are substituted for one tunnel, a long reservoir and an open channel. This substitution results in a modest cost increase. The long reservoir of Scheme III-A is discussed at length in the "Second Interim Report," and is considered, subject to further geological investigation, to have more advantages than disadvantages. With the same reservation, the tunnels of Scheme X appear to be in reasonably good ground and subject to no major hazards. Until geological investigations are more complete, it is not considered that either scheme has any advantage that would change the cost consideration in any way.

Scheme XI is the same as Scheme X except that it substitutes ridge route forebay and discharge pipes for Pastoria Creek forebay and discharge pipes. This substitution results in a modest additional cost increase. Although it appears that the first pumping plant for Scheme XI may be better founded than that for Schemes X and III-A, no reason is readily apparent for preferring the Scheme XI variation over the Scheme X variation.

Scheme XII is a two-unequal-lift ridge route arrangement utilizing a reservoir instead of the balancing tanks considered for Scheme V-A. It is somewhat more costly than Scheme V-A, and even more so relative to Scheme III-A. Some additional disadvantage is incurred by Scheme XII in that pumping heads are unequal, and therefore pump and motor spare parts are not interchangeable. The reservoir, though small, does have 25 minutes of 5,000 cfs flow in storage capacity between minimum and critical pump submergence levels. This gives some advantage over Scheme V-A, which has but 6 minutes storage, but a disadvantage compared with Scheme III-A, which has 75 minutes storage. The upper pumping plant for Scheme XII is located in a reservoir, as is that for Scheme III-A. Scheme V-A, with balancing tanks, is somewhat preferable in this respect over the other two schemes. Scheme XII utilizes Tunnel No. 3 to which some hazards in construction and safety are ascribed. So too does Scheme V-A; but Scheme III-A does not, thereby acquiring some advantage.

In summary, none of the three additional schemes considered in this supplement appear to have any substantial advantage or disadvantage over comparable schemes. The results of geological investigations now underway may be more significant than the relatively small cost differences now apparent.

CHAPTER IX

CONCLUSIONS AND RECOMMENDATIONS

The approximate differences between alternative schemes in construction costs, in present worth of average (using average of "maximum" and "minimum" assumed power rates) annual operating and maintenance costs and in the totals of these two costs are summarized below:

APPROXIMATE DIFFERENCES IN COST
Millions of Dollars

No. Lifts	Type of Pump	Pastoria Creek Sch.No.	Pastoria Creek Diff.	Ridge Sch.No.	Ridge Diff.
CONSTRUCTION COST DIFFERENCES					
1	4-stage, SS	-	-	VIII-A	+11.8
2	1-stage, SS	III-A	Base	V-A	+ 5.6
2	1-stage, SS	X	+ 3.4	XII	+11.4
2	1-stage, SS	XI*	+ 5.0	-	-
2	2-stage, DS-V	-	-	V-B	+13.3
2	2-stage, DS-H	-	-	V-C	+13.0
3	1-stage, SS	II-A	+12.5	IX	+20.5
PRESENT WORTH OF AVERAGE O & M COST DIFFERENCES					
1	4-stage, SS	-	-	VIII-A	+21.6
2	1-stage, SS	III-A	Base	V-A	+ 1.8
2	1-stage, SS	X	+ 1.2	XII	+ 0
2	1-stage, SS	XI*	+ 0.6	-	-
2	2-stage, DS-V	-	-	V-B	+18.0
2	2-stage, DS-H	-	-	V-C	+17.4
3	1-stage, SS	II-A	+ 3.0	IX	+ 4.2
TOTAL COST DIFFERENCES					
1	4-stage, SS	-	-	VIII-A	+33.4
2	1-stage, SS	III-A	Base	V-A	+ 7.4
2	1-stage, SS	X	+ 4.6	XII	+11.4
2	1-stage, SS	XI*	+ 5.6	-	-
2	2-stage, DS-V	-	-	V-B	+31.3
2	2-stage, DS-H	-	-	V-C	+30.4
3	1-stage, SS	II-A	+15.5	IX	+24.7

* Scheme XI is a combination of Pastoria Creek and Ridge routes.

Examination of the above table and consideration of the preceding discussion reveals no reason to make any change whatsoever in the "Conclusion and Recommendations" included in the "Second Interim Report" and starting on Page X-2.

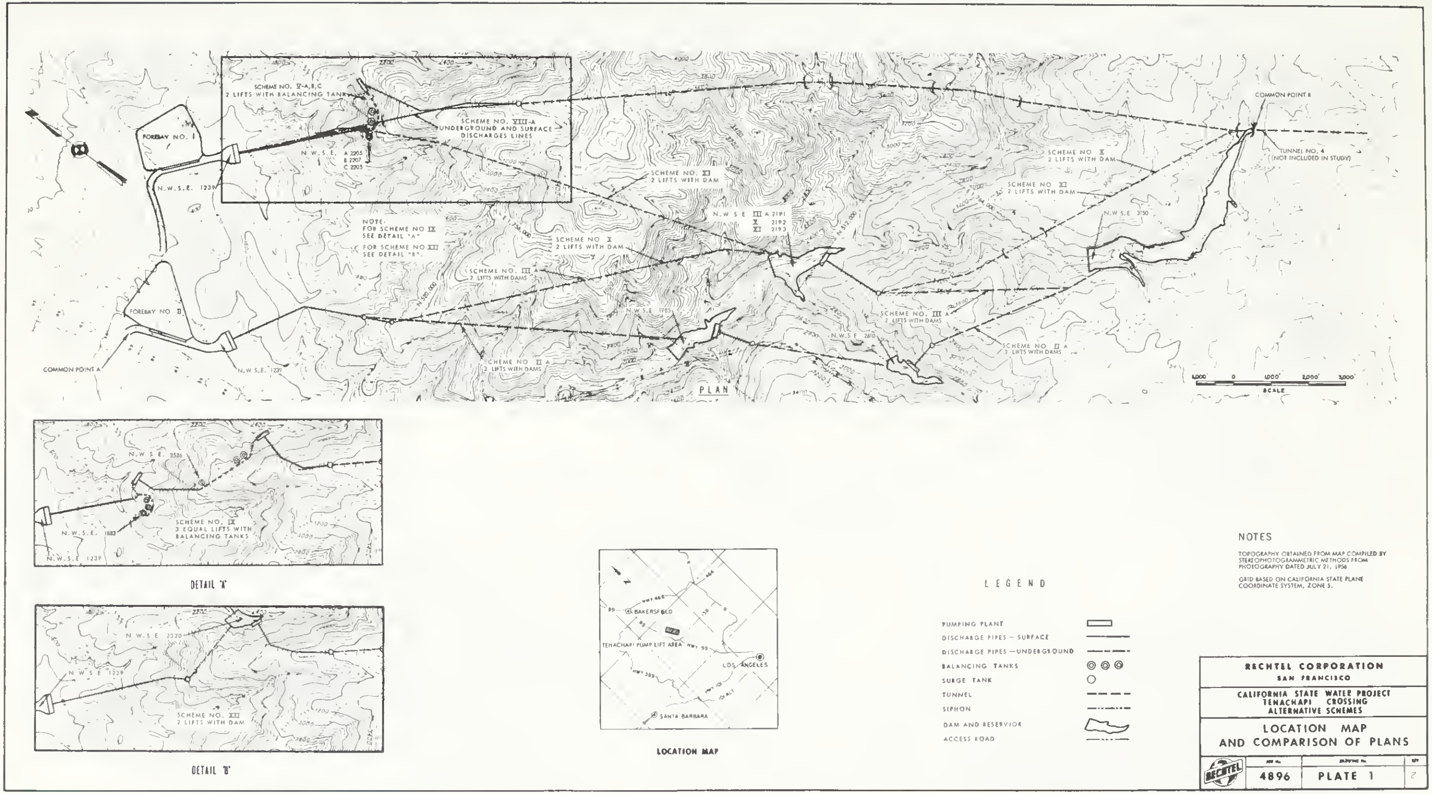
Consequently, and consistent with those "Conclusions and Recommendations", three schemes were selected for more refined analysis based upon a maximum capacity of 4,100 cfs, upon the lower power rates now considered to be available, and upon more complete geological investigations. These schemes are:

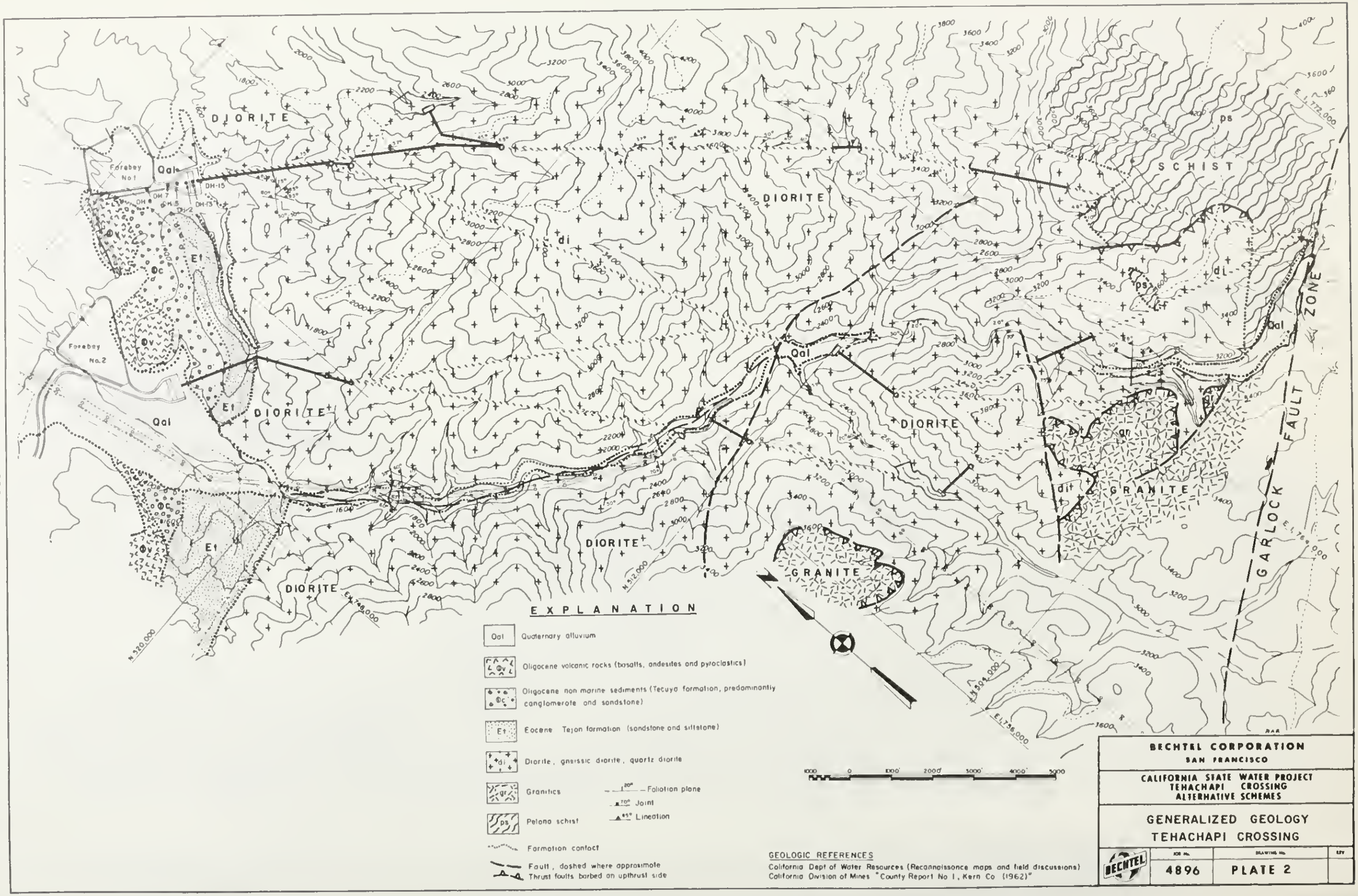
Scheme II - 3-lift, Pastoria Creek route

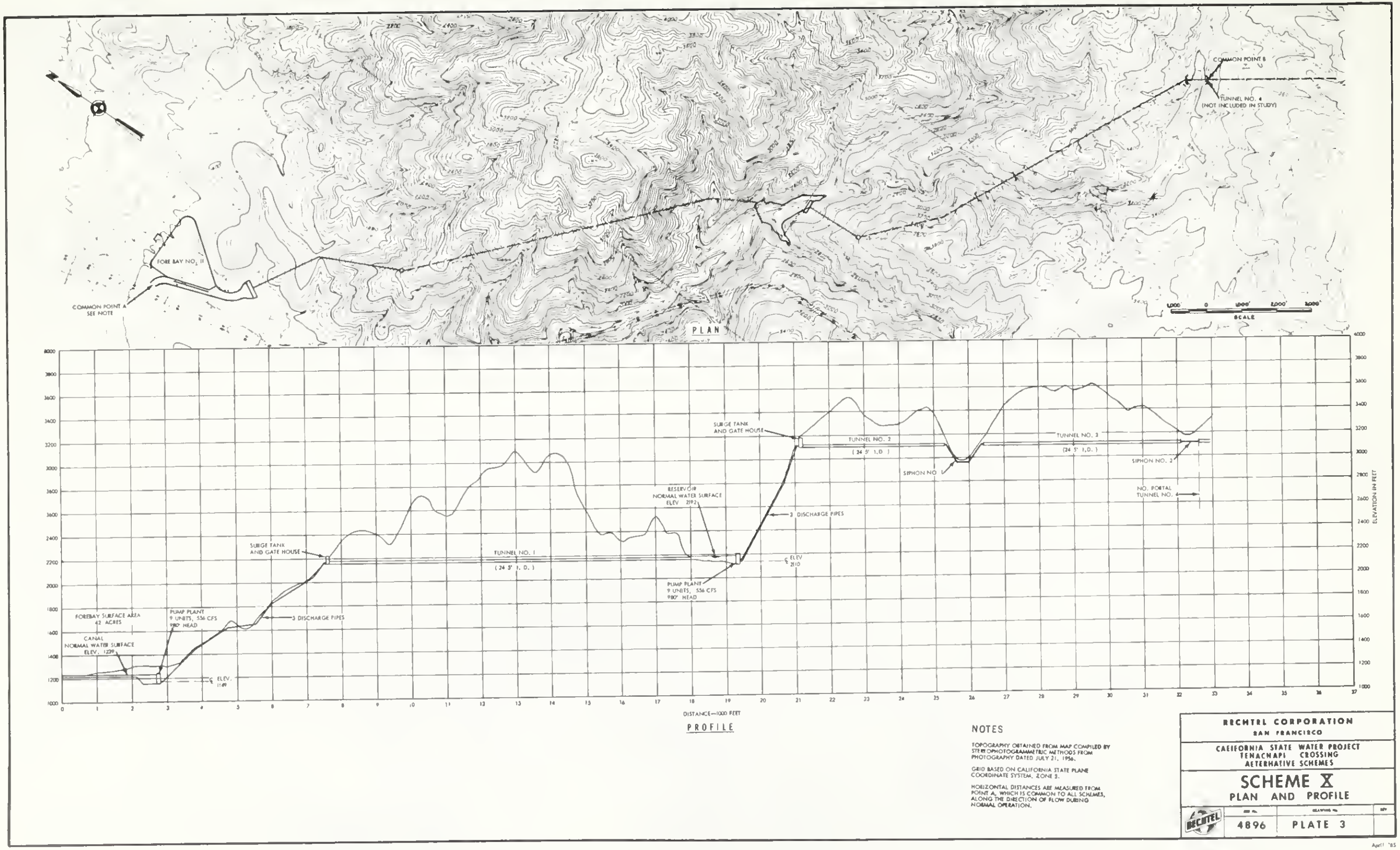
Scheme III - 2-equal-lift, Pastoria Creek route

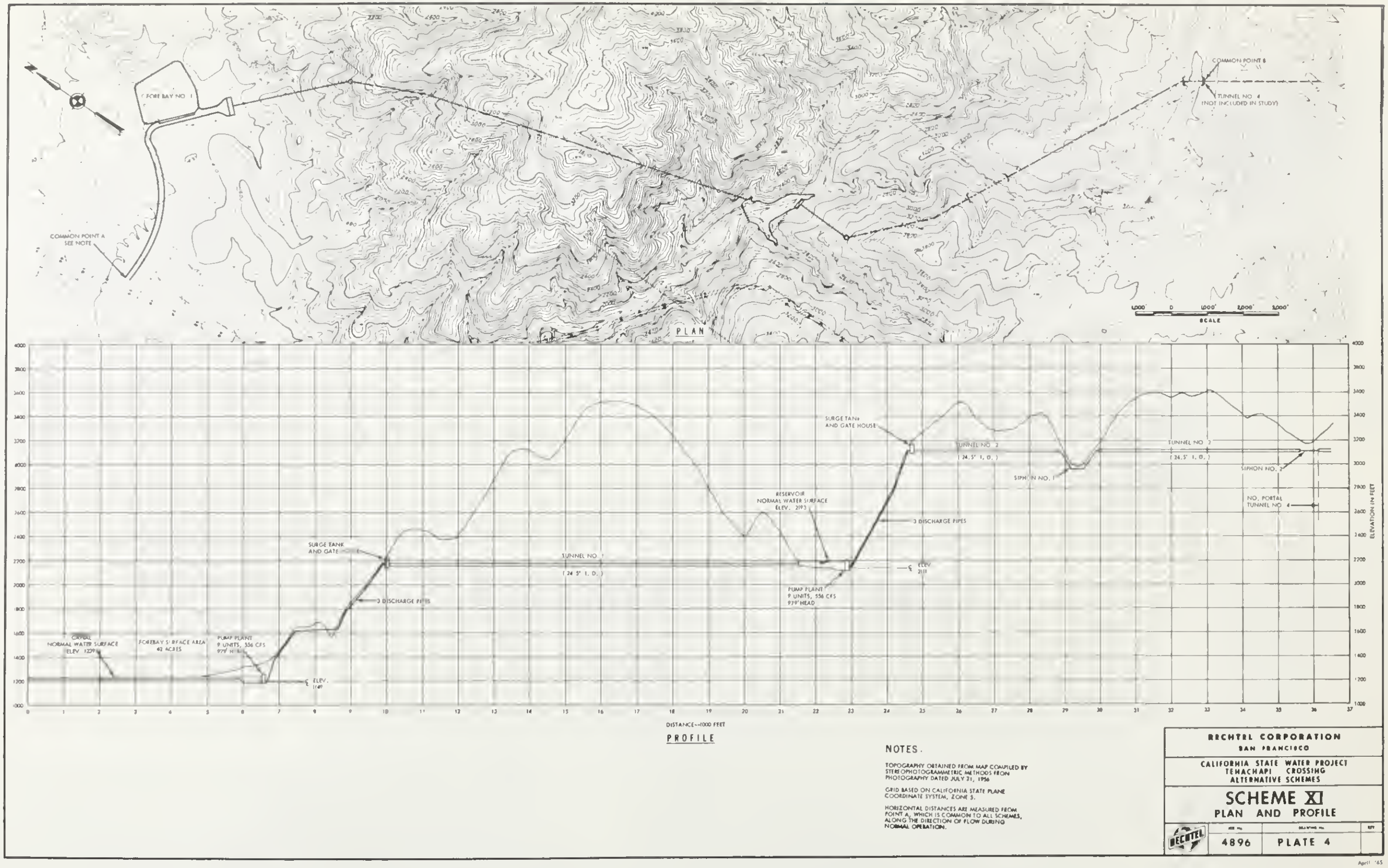
Scheme XI - 2-equal-lift, combination ridge and Pastoria Creek routes.

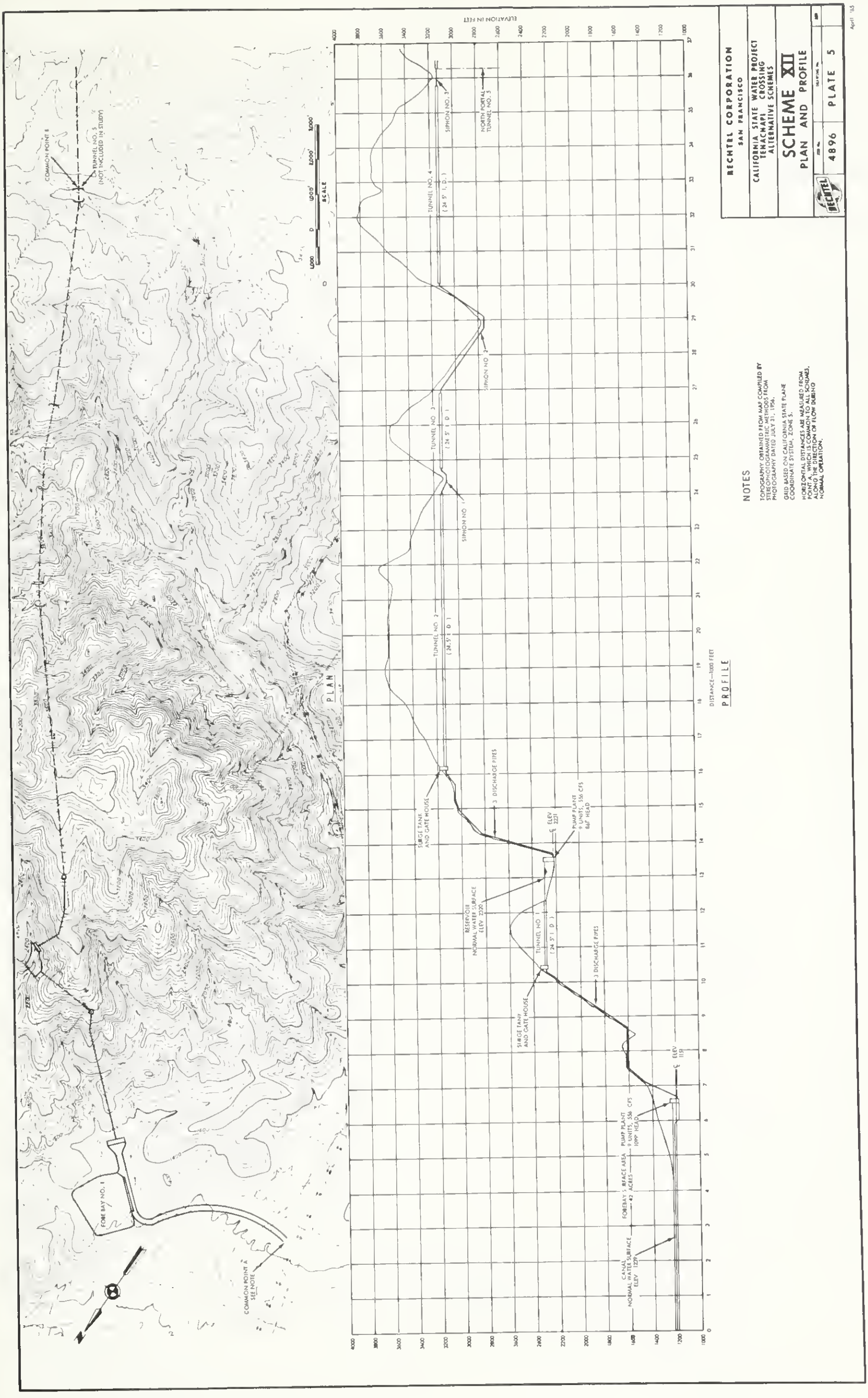
The studies are in progress, and early completion is expected.


FOREBAY NO. I
FOREBAY NO. II
COMMON POINT A
COMMON POINT B
N.W.S.E. 1239
SCHEME NO. V-A,B,C
2 LIFTS WITH BALANCING TANKS
N.W.S.E. A 2205 B 2207 C 2205
SCHEME NO. VIII-A
UNDERGROUND AND SURFACE
DISCHARGES LINES
SCHEME NO. XI
2 LIFTS WITH DAM
NOTE:
FOR SCHEME NO IX
SEE DETAIL "A"
FOR SCHEME NO XII
SEE DETAIL "B".
SCHEME NO X
2 LIFTS WITH DAM
SCHEME NO. III A
2 LIFTS WITH DAMS
SCHEME NO II A
3 LIFTS WITH DAMS
N.W.S.E III A 2191
X 2192
XI 2193
N.W.S.E 1983
N.W.S.E 2610
N.W.S.E 3150
TUNNEL NO. 4
(NOT INCLUDED IN STUDY)
PLAN
SCALE
DETAIL "A"
N.W.S.E. 2526
N.W.S.E. 1883
SCHEME NO. IX
3 EQUAL LIFTS WITH
BALANCING TANKS
DETAIL "B"
N.W.S.E 2320
SCHEME NO. XII
2 LIFTS WITH DAM
LOCATION MAP
BAKERSFIELD
TEHACHAPI PUMP LIFT AREA
LOS ANGELES
SANTA BARBARA
LEGEND
PUMPING PLANT
DISCHARGE PIPES — SURFACE
DISCHARGE PIPES — UNDERGROUND
BALANCING TANKS
SURGE TANK
TUNNEL
SIPHON
DAM AND RESERVIOR
ACCESS ROAD
NOTES
TOPOGRAPHY OBTAINED FROM MAP COMPILED BY STEREOPHOTOGRAMMETRIC METHODS FROM PHOTOGRAPHY DATED JULY 21, 1956
GRID BASED ON CALIFORNIA STATE PLANE COORDINATE SYSTEM, ZONE 5.
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
LOCATION MAP
AND COMPARISON OF PLANS
4896
PLATE 1


DIORITE
SCHIST
ps
di
Qal
Et
Φv
Φc
GRANITE
gr
GARLOCK FAULT ZONE
Forebay No 1
Forebay No. 2
DH-15
EXPLANATION
Qal Quaternary alluvium
Oligocene volcanic rocks (basalts, andesites and pyroclastics)
Oligocene non marine sediments (Tecuya formation, predominantly conglomerate and sandstone)
Et Eocene Tejon formation (sandstone and siltstone)
di Diorite, gneissic diorite, quartz diorite
gr Granitics
ps Pelona schist
Formation contact
Fault, dashed where approximate
Thrust faults barbed on upthrust side
Foliation plane
Joint
Lineation
GEOLOGIC REFERENCES
California Dept of Water Resources (Reconnaissance maps and field discussions)
California Division of Mines "County Report No 1, Kern Co (1962)"
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
GENERALIZED GEOLOGY
TEHACHAPI CROSSING
4896
PLATE 2


COMMON POINT B
TUNNEL NO. 4
(NOT INCLUDED IN STUDY)
FORE BAY NO. II
COMMON POINT A
SEE NOTE
SCALE
PLAN
SURGE TANK
AND GATE HOUSE
TUNNEL NO. 2
(24' 5" I.D.)
TUNNEL NO. 3
(24' 5" I.D.)
SIPHON NO. 1
SIPHON NO. 2
NO. PORTAL
TUNNEL NO. 4
RESERVOIR
NORMAL WATER SURFACE
ELEV. 2192
3 DISCHARGE PIPES
SURGE TANK
AND GATE HOUSE
TUNNEL NO. 1
(24' 5" I.D.)
ELEV.
2110
PUMP PLANT
9 UNITS, 556 CFS
980' HEAD
FOREBAY SURFACE AREA
42 ACRES
CANAL
NORMAL WATER SURFACE
ELEV. 1239
PUMP PLANT
9 UNITS, 556 CFS
980' HEAD
3 DISCHARGE PIPES
ELEV.
1149
ELEVATION IN FEET
DISTANCE—1000 FEET
PROFILE
NOTES
TOPOGRAPHY OBTAINED FROM MAP COMPILED BY STEREOPHOTOGRAMMETRIC METHODS FROM PHOTOGRAPHY DATED JULY 21, 1956.
GRID BASED ON CALIFORNIA STATE PLANE COORDINATE SYSTEM, ZONE 5.
HORIZONTAL DISTANCES ARE MEASURED FROM POINT A, WHICH IS COMMON TO ALL SCHEMES, ALONG THE DIRECTION OF FLOW DURING NORMAL OPERATION.
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TENACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME X
PLAN AND PROFILE
4896
PLATE 3
April '65


COMMON POINT B
TUNNEL NO. 4
(NOT INCLUDED IN STUDY)
FORE BAY NO. 1
COMMON POINT A
SEE NOTE
1,000' 0 1,000' 2,000' 3,000'
SCALE
PLAN
SURGE TANK
AND GATE HOUSE
TUNNEL NO. 2
(24.5' I. D.)
SIPHON NO. 1
SIPHON NO. 2
NO. PORTAL
TUNNEL NO. 4
RESERVOIR
NORMAL WATER SURFACE
ELEV. 2193
3 DISCHARGE PIPES
SURGE TANK
AND GATE
TUNNEL NO. 1
(24.5' I. D.)
PUMP PLANT
9 UNITS, 556 CFS
979' HEAD
3 DISCHARGE PIPES
CANAL
NORMAL WATER SURFACE
ELEV. 1229
FOREBAY SURFACE AREA
42 ACRES
PUMP PLANT
9 UNITS, 556 CFS
979' H.
ELEV. 1149
ELEVATION IN FEET
DISTANCE--1000 FEET
PROFILE
NOTES.
TOPOGRAPHY OBTAINED FROM MAP COMPILED BY STEREOPHOTOGRAMMETRIC METHODS FROM PHOTOGRAPHY DATED JULY 31, 1956
GRID BASED ON CALIFORNIA STATE PLANE COORDINATE SYSTEM, ZONE 5.
HORIZONTAL DISTANCES ARE MEASURED FROM POINT A, WHICH IS COMMON TO ALL SCHEMES, ALONG THE DIRECTION OF FLOW DURING NORMAL OPERATION.
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME XI
PLAN AND PROFILE
BECHTEL
4896
PLATE 4
April '65


BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME XII
PLAN AND PROFILE
4896
PLATE 5
April '65
NOTES
TOPOGRAPHY OBTAINED FROM MAP COMPILED BY STEREOPHOTOGRAMMETRIC METHODS FROM PHOTOGRAPHY DATED JULY 31, 1956.
GRID BASED ON CALIFORNIA STATE PLANE COORDINATE SYSTEM, ZONE 5.
HORIZONTAL DISTANCES ARE MEASURED FROM POINT A, WHICH IS COMMON TO ALL SCHEMES, ALONG THE DIRECTION OF FLOW DURING NORMAL OPERATION.
PLAN
PROFILE
DISTANCE—1000 FEET
ELEVATION IN FEET
SCALE
COMMON POINT A SEE NOTE
COMMON POINT B
TUNNEL NO. 5 (NOT INCLUDED IN STUDY)
FORE BAY NO. 1
CANAL NORMAL WATER SURFACE ELEV 1229
FOREBAY SURFACE AREA 42 ACRES
PUMP PLANT 9 UNITS, 556 CFS 1099' HEAD
ELEV 1151
SURGE TANK AND GATE HOUSE
3 DISCHARGE PIPES
TUNNEL NO. 1
(24' 5" I. D.)
RESERVOIR NORMAL WATER SURFACE ELEV 2320
ELEV 2231
PUMP PLANT 9 UNITS, 556 CFS 868' HEAD
SURGE TANK AND GATE HOUSE
3 DISCHARGE PIPES
TUNNEL NO. 2
(24' 5" I. D.)
SIPHON NO. 1
TUNNEL NO. 3
(24' 5" I. D.)
SIPHON NO. 2
TUNNEL NO. 4
(24' 5" I. D.)
SIPHON NO. 3
NORTH PORTAL TUNNEL NO. 5

GEOLOGIC PROGRESS REPORT NO. 2
ON
ALTERNATIVE SCHEMES
FOR THE
TEHACHAPI CROSSING
OF THE
CALIFORNIA STATE WATER PROJECT

FOR

THE METROPOLITAN WATER DISTRICT
OF SOUTHERN CALIFORNIA

BECHTEL CORPORATION
San Francisco
April 30, 1965

GEOLOGIC PROGRESS REPORT NO. 2
ALTERNATIVE SCHEMES
TEHACHAPI CROSSING
CALIFORNIA STATE WATER PROJECT

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GEOLOGIC PROGRESS REPORT NO. 2
ALTERNATIVE SCHEMES
TEHACHAPI CROSSING
CALIFORNIA STATE WATER PROJECT

INTRODUCTION

Since issuing the progress report "The Evaluation of Geologic Conditions for Alternate Schemes on the Tehachapi Crossing Project", April 1, 1965, additional geologic investigation of the Tehachapi Pump Lift schemes has been conducted by Bechtel geologists. This additional investigation consisted of:

1. Geologic reconnaissance of forebay sites, tunnel alignments, discharge lines, siphons and the ridge route dam site.
2. Reconnaissance geologic mapping of rock and overburden conditions of the four proposed Pastoria Creek dam sites, reservoir areas and pumping plants.
3. Core drilling at the axis of the upstream dam site.
4. Conferences with geologists of the California Division of Water Resources.
5. Evaluation of some of the exploration completed by the California Division of Water Resources.

Part A of this report summarizes the geologic conditions at the four proposed Pastoria Creek dam sites. Geologic maps and sections of dam sites, pumping plants, and reservoir areas are enclosed with this report. Due to the limited amount of time for this work and the absence of survey control, the enclosed maps are incomplete and the locations of geologic features are only of sketch accuracy. Rock type names used in this report are based on field identification.

Part B of this report summarizes geologic conditions along the proposed alignments of Schemes X-A, II-B, III-B, XI-A and XII-A. Geologic inspection of these alignments was a reconnaissance.

CONCLUSIONS AND RECOMMENDATIONS

The following are our opinions based on geologic investigations of the Tehachapi Pump Lift area.

1. Geologic conditions in Pastoria Creek are adequate for constructing any of the proposed two- or three-lift scheme dam sites. Some of these conditions are not ideal and more test data on materials in certain areas are necessary to determine proper design criteria.

2. The two downstream sites of the three-lift scheme (Scheme II-B) surficially appear to be the best of the proposed Pastoria Creek dam sites due to less stripping and more competent foundations. These two sites should be explored to determine subsurface conditions.

3. Exploration at the uppermost Pastoria Creek dam site is still in progress. This exploration may show that foundation conditions at the proposed downstream alternate axis are as good as the foundation conditions at the two lower dam sites of the three-lift scheme.

4. The proposed ridge route two-lift scheme dam site (Scheme XII-A) appears to be underlain by competent rock.

5. There is a geologic advantage of tunnel conditions along the Pastoria Creek alignment over tunnel conditions along the ridge alignment. The uppermost Pastoria Creek dam site replaces the third section of tunnel (DWR Tunnel No. 3) which is assumed to require heavier support and attaches more uncertainties to the effective operation of the pump lift across this faulted area.

6. From a standpoint of safety and pump lift integrity, potential hazzards to tunnels and to dam sites appear equal even though each type of structure presents a somewhat different set of problems.

7. All of the proposed structures in the Tehachapi Pump Lift area are in a seismically active region.

8. A portion of the Garlock fault zone is exposed in the excavation for the pilot tunnel of Tunnel No. 4 on the DWR program. This is an important fault and one along which movement has displaced alluvium, presumably of Recent geologic age.

PART A

GEOLOGY OF THE PASTORIA CREEK DAM SITES

A.1 General Geology. Rocks in the general Pastoria Creek area being considered as possible dam sites for the Tehachapi Pump Lift are extremely variable both in lithology and in physical properties. Gneissoid granitic intrusions and metasediments occur throughout the area. Nearly all of these rocks appear to have been subjected to several periods of faulting and to varying degrees of metamorphism. Weathering has softened some of these rocks to depths probably greater than 200 feet.

A.2 Seismicity. All of the dam sites are located in a very active seismic area. The northeast trending White Wolf fault lies about 11 miles to the north; the San Andreas fault lies about 6 miles to the south; the Garlock fault cuts through the reservoir of the upper dam site. (See Plate 11.) All of these faults are considered major structural features of Southern California and all but the Garlock fault are known to have moved during recorded history. Many smaller faults occur throughout the area which includes the proposed dam sites, but evidence of recent movement along them has not been found.

A.3 Scheme II-B, Dam Site No. 1. Dam Site No. 1 and reservoir are underlain by fractured gneissoid granitic rocks and metasediments which generally range from moderately firm to very firm. Slope wash covers most of the valley walls and alluvium occurs in the channel section. Older terrace deposits occur between elevations 2000 and 2150, upstream of the proposed axis along the right bank of the reservoir. (See Plate 9.)

A.3.1 Foundation. An average of 15 feet of slope wash will require stripping on the left abutment; but where exposed, the rocks are moderately firm to very firm and only minimum trimming of the upper surface of bedrock will be necessary.

The channel section contains about 30 feet of alluvium which will require stripping. The maximum thickness occurs along the left side of this section where slope wash is intermixed with the alluvium. (See Plate 1.)

Rock crops out over most of the right abutment below elevation 2000. The upper surface of rock exposed near the valley floor is oxidized brown, thinly foliated, friable and moderately firm. It is estimated that an average stripping of 5 feet should provide an adequate foundation for an earth or rock fill dam.

Fractures on both abutments and in the channel section are estimated to be open to ground water percolation for depths from 50 to 100 feet. Grouting should improve the fractured rock.

A.3.2 Pumping Plant. The pumping plant will be founded on sound bedrock after maximum stripping of about 15 feet of alluvium and slope wash. (See Plate 2.) The discharge line will be in the same general type of rock, but weathering probably becomes progressively deeper higher on the hillside. Above elevation 3000 weathering may be 50 to more than 100 feet deep.

A.3.3 Reservoir. No adverse geologic conditions were found in the reservoir. Overburden in areas of creep is shallow and no major sliding during saturation and drawdown is anticipated.

A.4 Scheme III-B, Dam Site No. 1. Rocks underlying this dam and reservoir are variable. There is at least two different types of intrusives and a wedge of metasediments trending through the area. A rock, field-classified as hornblende gneiss, forms the right abutment. The left abutment and upstream right bank of the reservoir contain both metasediments and gneissoid granitic rocks. The metasediments include a 15 foot thick bed of metamorphosed limestone which, within a distance of about 400 feet, changes in attitude from N 58 E, 25 SE to N 60 E, 60 NW. A gray-black rock, field-classified as gneissic diorite, occurs upstream of these rocks. (See Plate 3.)

Small shear zones are common in all of these rock types but the block of metasediments appears to have undergone the greatest degree of contortion. This may have resulted from intrusion of the gneissic diorite into the metasediments and hornblende gneiss.

A.4.1 Foundation. The left abutment is formed by a deeply weathered and terraced nose of gneissoid granitic and metasedimentary rocks. Several 1 to 2 foot wide, northwest trending shear zones cross this abutment, but these shear zones do not differ appreciably from adjacent soft to moderately firm rock where exposed in dozer road cut-slopes.

It is estimated that 15 feet of stripping will provide an adequate foundation for an earth fill dam.

An average of 30 feet of alluvium and slope wash will require stripping in the channel section.

In general the right abutment is underlain by firm rock except in the upstream area where closely foliated zones of moderately firm to friable material occurs. About 5 feet of stripping over the right abutment will be necessary.

It is assumed that the left abutment has a low permeability due to the decomposition of the rock and the silt and clay filling of the fractures. Across the channel section and on the right abutment, where the rock is more brittle, fractures are estimated to be open 50 to 100 feet deep.

A.4.2 Pumping Plant. The pumping plant is located on the upstream left bank in a topographic amphitheater which was presumably formed in part by the former meandering of Pastoria Creek and in part by creep of slope wash from the above steep hillside. Surficially the area appears unstable. Some of the overburden may require stripping prior to construction of the pumping plant. It is estimated that overburden averages 25 feet deep in the pumping plant area and about

70 feet deep along the discharge pipe alignment to elevation 2400. Above elevation 2400 the overburden is assumed to be shallowing until at elevation 2700 it is probably less than 15 feet deep. Sampling and testing is necessary to determine how much of this overburden will require stripping to ensure stability in the pumping plant area.

DWR drilling shows that underlying bedrock is closely fractured and moderately firm to soft gneissoid granitic and metasedimentary rocks. (See Plate 4.)

A.4.3 Reservoir. The unstable area at the pumping plant could be the most important consideration related to the integrity of this dam and reservoir. The upper surface of the overburden is presently creeping down-slope. Depending on the exact subsurface configuration of the bedrock and the shearing strength of this overburden, a deep cut in the toe area could initiate large scale land-sliding.

Areas of shallow slope failure were also found in the contorted block of metasediment rock along the upstream right bank of the reservoir. However, none of these are of sufficient size or so situated to cause sudden sliding of a large mass of material in the reservoir.

A.5 Scheme II-B, Dam Site No. 2. The proposed site of Dam No. 2 is situated in a rather straight, narrow canyon carved by stream action along the foliation strike of a firm, gray black gneissic diorite. The left canyon wall consists of widely scattered rock outcrops, numerous zones of boulder float rock and loosely jointed bedrock, and creeping overburden which in places may be as much as 40 feet deep. These conditions along the left canyon wall are mainly due to the attitude of the foliation. The foliation generally dips into the canyon at 20 to 30 degrees. More continuous rock outcrops occur lower on the left canyon wall, near the edge of the channel section.

The right canyon wall is composed of rugged rock outcrops which are oxidized to depths ranging from 5 to more than 30 feet. The foliation dips into the hill at 20 to 30 degrees. Alternating massive and closely foliated layers coupled with the attitude of the foliation has caused undersapping of some of the more massive layers. Resulting shallow rock slides have occurred in some areas. (See Plate 9.)

A.5.1 Foundation. The left abutment is composed of firm but closely fractured gneissic diorite overlain by slope wash. It is estimated that a combined stripping of slope wash and loose jointed rock to a depth of 15 feet will be required for an earth or rock fill dam.

The channel section is estimated to contain 10 or 15 feet of slope wash and alluvium which will require stripping.

The right abutment is underlain by nearly continuous rock outcrop above elevation 2500. Depending on the precise location of the shallow rock slide above elevation 2800 on the right abutment, a crib wall or some additional stripping may be necessary at higher elevations to insure stability of this material during construction. (See Plate No. 5.) It is estimated that an average of about 5 feet of stripping will be required on the right abutment.

Fractures may be open to depths ranging from 50 to 100 feet on the right and left abutments and probably to 50 feet deep across the channel section. Some grouting of these open fractures will probably be necessary.

A.5.2 <u>Pumping Plant.</u> The pumping plant should be in bedrock after stripping a maximum of 15 feet of slope wash and alluvium. (See Plate 6.) Moderate weathering will probably extend for 10 to 15 feet into bedrock at this site. The discharge pipe alignment has rock outcrops and locally as much as 15 feet of slope wash between elevations 2900 and 3200. Rocks along this alignment are estimated to be sound beneath a 30 foot zone of surface weathering.

The left canyon wall was inspected to determine the possibility of landsliding of the deeper overburden at higher elevations into the pumping plant structure during pool saturation and drawdown. (See Plate 6.) Several bedrock exposures were found at about elevation 2540 to 2580, across the canyon from the pumping plant. This bedrock should provide adequate toe support for the deeper overburden at higher elevations during pool saturation and drawdown. Some exploration would aid in evaluating these conditions.

A.5.3 <u>Reservoir.</u> Several areas of creep and probable deep overburden are suspected on the left bank of the reservoir. The deepest overburden area of this type appears to be on the left bank near the end of pool where creeping overburden is estimated to be from 20 to 40 feet deep. Scattered bedrock exposures near the left abutment valley floor suggest there is adequate support at the toe of these deeper overburden areas to prevent large scale sliding into the reservoir. Some exploration should be done to aid in determining the exact depth of the overburden in some of these areas.

A.6 <u>Scheme II-B, Dam Site No. 3; or Scheme III-B, Dam Site No. 2, (Common Sites).</u> Rock types in this dam and reservoir area are exceedingly variable. Much of this complexity is assumed due to the proximity of this site to the

Garlock fault zone. The site is underlain by gneissoid granitics, granites and metasediments including metamorphosed limestone, phyllites, schists, and a reddish breccia of unknown origin. These range in physical properties from very sound and massively jointed rocks to decomposed and crushed soil-like material.

The Garlock fault zone and several smaller shear zones cut the rocks at this site. The exact limits of the Garlock fault zone is not readily apparent but a northern limit to the main branch has been tentatively determined. (See Plate 10.) This determination is based on exposures of the Garlock fault zone in the excavation for the portal of Tunnel No. 4 and in nearby road cuts along the road leading into this area from Lebec, California. The main fault zone is believed to trend out of the reservoir area about 2800 feet downstream of the portal of the pilot tunnel for DWR Tunnel No. 4. This point is about 4000 feet upstream from the proposed dam axis shown on Plate 10.

A probable thrust fault was mapped in a side canyon about 1300 feet upstream of the axis. This is exposed along an old dozer road which has a 200 foot long cut bank composed of intermixed sections of slope wash and broken rock with up to 1 foot thick sections of low angle gravelly clay gouge.

The thrust zone appears to be about 20 feet thick measured normal to the fault surface. The precise attitude of this thrust could not be determined. Whether or not it projects into the dam abutments may be determined by the drilling.

An alternate axis is also proposed about 1000 feet downstream from the axis presently being drilled. Surficially the abutment rocks at the alternate axis appear more competent. (See Plate 8 and 10.) This axis is presently being explored.

A.6.1 Foundation. The foundation is underlain by gneissoid granitic rocks and metasediments which include the reddish breccia.

The general foundation conditions are poor. Shearing, hydrothermal alteration and deep weathering have reduced much of the rock to residual soil. Depths to firm foundation rock are probably too great to consider stripping the abutments. The residual rock must be thoroughly sampled and tested to determine the slopes required for an earth dam on such material.

On the right abutment, sound rock ribs occur adjacent to weathered and hydrothermally altered rock which is practically a residual soil for a depth of more than 60 feet at places. More information is required to determine a suitable stripping depth, but it is tentatively assumed for estimating purposes that stripping the upper 5 to 10 feet of slope wash and a few feet of the severely weathered bedrock will provide a suitable earth fill foundation. For final design, sampling and testing are required to determine the (1) density, (2) shearing strength, (3) nature of possible clayey slip planes, and (4) subsurface configuration of hard and severely weathered bedrock both above and below the proposed crest elevation. A resistant 15 to 20 foot wide rib of bedrock occurs from 10 to 50 feet upstream of the proposed axis which will decrease stripping depths in the upstream area. A rib of resistant red breccia, presumably produced by silicification along a fault, occurs about 300 feet downstream of the axis. The dam axis could be oriented to take advantage of these two rock ribs. (See Plate 7 and 10.)

The channel section contains about 35 feet of alluvium and slope wash which should be excavated for the zone 1 material or core trench. Samples and tests are needed to determine the nature of the channel deposits.

The left abutment is underlain by closely fractured and weathered gneissoid granitic rocks and 10 to 15 feet of slope wash. It is estimated that 15 feet of stripping will be required on this abutment.

A.6.2 Reservoir. The reservoir was mapped to determine overburden conditions which might produce sliding into the reservoir during saturation and drawdown. Several areas of creep with from 10 to 30 feet of overburden occur on the left bank about 2200 to 3600 feet upstream of the proposed axis. (See Plate 10.) A computation of the estimated maximum volume of the overburden in these areas was made to determine the effect which would be produced if they did slide into the reservoir. Based on this computation, none are of sufficient size to produce waves which would overtop the dam.

PART B

GEOLOGY OF SCHEMES X-A, II-B, III-B, XI-A and XII-A

B.1 General Geology. Rocks occurring along the alignments of Schemes X-A, II-B, III-B, XI-A and XII-A are generally the same as those described for the four Pastoria Creek dam sites under Part A of this report. However, the forebay areas, first-lift pumping plants, and portions of the first-lift discharge pipe alignments are underlain by younger rocks with different physical properties.

These younger strata are (1) volcanic rocks of either Miocene or Oligocene age, (2) the Tecuya formation of

Oligocene age, and (3) the Tejon formation of Eocene age. The volcanic rocks are sound lava flows and soft to firm flow breccias. The Tecuya formation is composed predominantly of continental sandstone and conglomerate, some of which appear tuffaceous. The marine Tejon formation is predominantly poorly cemented to friable sandstones.

The Tertiary formations have been tilted to the north presumably during uplift of the Tehachapi Mountains. They dip irregularly northerly at about 30°. No major faults were found in these Tertiary strata.

B.2 <u>Seismicity.</u> The epicenter of the 1952 Arvin-Tehachapi earthquake is located about 11 miles to the north-west of the Tertiary strata and alluvial deposits which occur in the forebay areas. (See Plate 11.) This earthquake, which had a magnitude of 7.7 on the Richter scale, is believed to be the result of movement along the White Wolf fault, (Dibblee, 1955, Division of Mines Bulletin 171).

B.3 <u>Scheme X-A.</u> This scheme is shown on Plate 12. The second-lift of this scheme is the same as the second lift of Scheme III-B, which is described under Section A.4 of this report.

B.3.1 <u>Forebay, First Pump Lift, and Tunnel.</u> The forebay is underlain by poorly to unconsolidated Recent alluvium and moderately consolidated sediments. The lava which occurs locally in this area will probably underlie portions of the forebay.

All excavation in these Tertiary rocks will be common and cut slopes should be laid back at least 1:1. Cut slopes in the alluvium may require flattening to 2:1.

The pumping plant will probably be founded on the Tecuya formation which should have adequate bearing strength to support the structure.

The portion of the discharge pipe above approximately elevation 1600 will be in gneissoid granitic rocks. Surface weathering is estimated to extend more than 50 feet deep.

Outcrops exposed over the alignment of the tunnel to the dam site are predominantly moderately firm to very firm gneissoid granitic rocks. Severe weathering is probably less than 50 feet and this section of tunnel should require only light support.

B.3.2 Second Pump Lift and Tunnels. Pumping plant conditions for the second lift dam site are described in Section A.5.2. Beyond the unstable area described in Section A.5.2 the discharge pipe alignment crosses moderately weathered, hard gneissoid granitic rocks overlain by less than 15 feet of overburden.

The tunnel alignment from the second-lift surge tank to the siphon crossing Pastoria Creek passes through an area of hard gneissoid granitic rocks where only light tunnel support is anticipated.

On the east side of the siphon, rocks appear to be weathered to depths in excess of 100 feet. The tunnel through these gneissoid granitic rocks and metasediments will probably require light to moderately heavy support. As tunnelling progresses into the schistose metasediments affected by the Garlock fault, moderately heavy to heavy tunnel support may be required.

B.4 Schemes III-B and II-B. Geologic conditions along the alignments of Schemes II-B and III-B are similar to the geologic conditions of Scheme X-A. However, the uppermost Pastoria Creek dam site eliminates the siphon across Pastoria Creek and the upper tunnel common to Schemes X-A and XI-A.

B.5 Scheme XI-A. Scheme XI-A is shown on Plate 12. The second-lift of this scheme is the same as Scheme X-A. It is described in Sections B.3.2 and A.4 of this report.

B.5.1 Forebay, First Pump Lift and Tunnel. The proposed forebay of Scheme XI-A is underlain by the same Tertiary strata described in Scheme X-A. In general the engineering considerations should be the same for both forebays.

The upper portion of the pumping plant excavation will be in the alluvium, but the lower part should be in gneissoid granitic rocks. Cut slopes of 2:1 in the alluvium should be stable and the gneissoid granitic rocks may stand at $\frac{1}{2}$:1.

The discharge pipe alignment of this first pump lift is in gneissoid granitic rocks which are estimated to be weathered, closely fractured, and soft to depths in excess of 75 feet.

Nearly continuous exposures of hard gneissoid granitic rocks occur along the tunnel alignment from the discharge pipe to the second-lift dam site. The rock should be hard and only light tunnel support is anticipated.

B.6 Scheme XII-A. The first pump lift of Scheme XII-A is the same as for the Scheme XI-A up to the surge tank at about elevation 2300. From this surge tank a tunnel runs easterly directly to a reservoir at about elevation 2320. (See Plate 2.)

Outcrops along this tunnel alignment are moderately firm to very firm gneissoid granitic rock which may be weathered to 50 feet deep. Only light tunnel support is anticipated.

B.6.1 Second Pump Lift and Tunnels. The dam site is underlain by sound gneissoid granitic rocks. There appears to be from 10 to 20 feet of slope wash and weathered rock along the proposed axis. Drilling by DWR approximately 900 feet downstream of this site suggests that open fractures which may require grouting extend for 40 to 50 feet in depth.

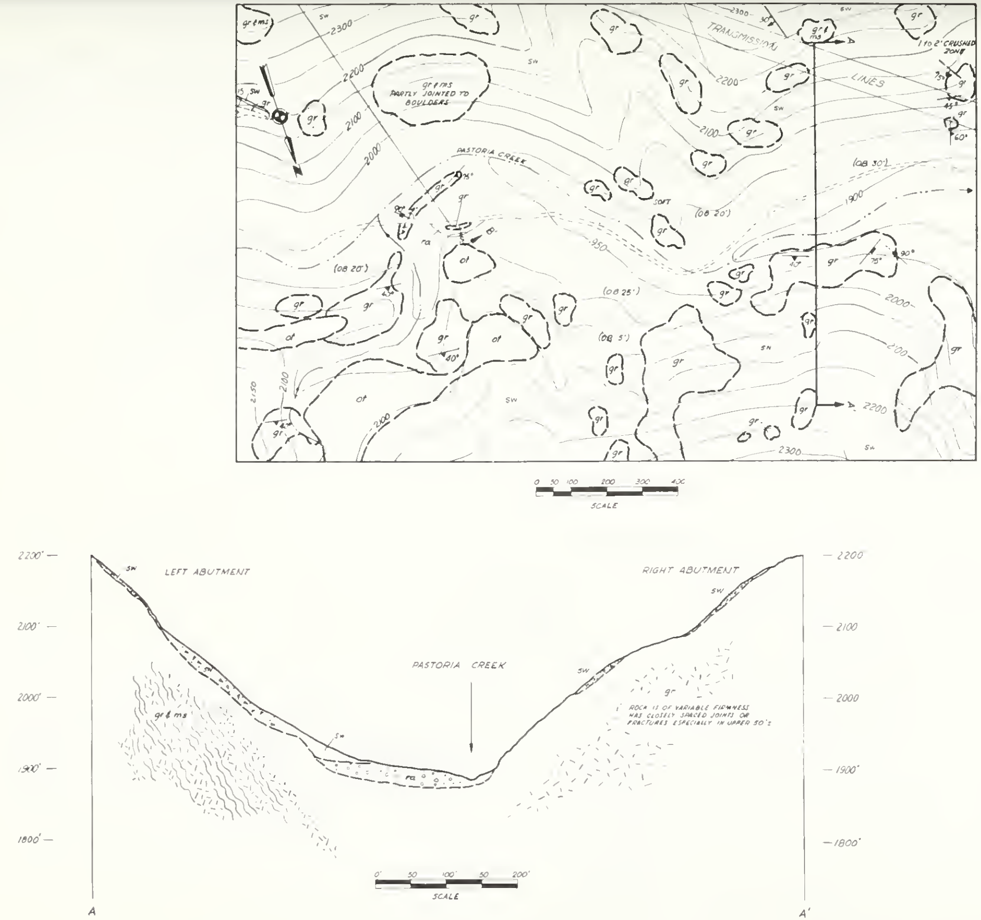
Conditions at the proposed pumping plant and discharge line are expected to be about the same as for the dam. Rock is expected to have less overburden and less weathering above elevation 2800 to the surge tank.

Outcrops along the alignment of the tunnel from the surge tank to the first siphon are hard jointed gneissoid granitic rocks. It is estimated that only light tunnel support will be required.

Conditions along the alignment between the second tunnel and second siphon are similar to the first tunnel. Only light support is anticipated except in the southern part of the tunnel where poorer rock conditions are expected.

Both siphon areas are deeply weathered. Drilling by DWR shows that severe weathering may extend for more than 150 feet deep in the second siphon area where serious construction problems are anticipated. Exploration dozer trenches by DWR show that slope wash is shallow but that the upper portion of bedrock has been weathered to soil.

Rocks across the second siphon and along the third tunnel alignment are deeply weathered gneissoid granitic and schistose metasedimentary rocks which may require moderately heavy support. As tunnelling progresses into the area affected by the Garlock fault zone, moderately heavy to heavy support is anticipated.



EXPLANATION

- SLOPE WASH — ANGULAR ROCK FRAGMENTS, SAND, SILT, AND/OR CLAY, UNCONSOLIDATED
- RECENT ALLUVIUM — GRAVEL, SAND, AND SILT, UNCONSOLIDATED
- OLDER TERRACE DEPOSITS — GRAVEL, SAND, SILT AND/OR CLAY, POORLY CONSOLIDATED
- GRANITIC ROCKS — INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS, UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
- METASEDIMENTS AND GRANITIC ROCKS — INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES, SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE, SCATTERED SMALL SHEAR ZONES
- BRECCIA — REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
- STRIKE AND DIP OF FOLIATION, VERTICAL 90°
- STRIKE AND DIP OF JOINTS, VERTICAL 90°
- STRIKE AND DIP OF BEDDING, VERTICAL 90°
- STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
- STRIKE AND DIP OF THRUST FAULT DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
- AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
- APPROXIMATE GEOLOGIC CONTACT
- VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN DDH - DRILLED BY DWR DH - DRILLED BY BECHTEL CORP
- CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
- SLOPE ANGLE
- UNIMPROVED ROAD OR DOZER TRAIL
- CREEK OR INTERMITTENT STREAM
- TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL

NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10

GEOLOGY BY H C FIFER AND C L TRANTHAM 4-16-65

DRAWN BY R F SORIANO 4-28-65

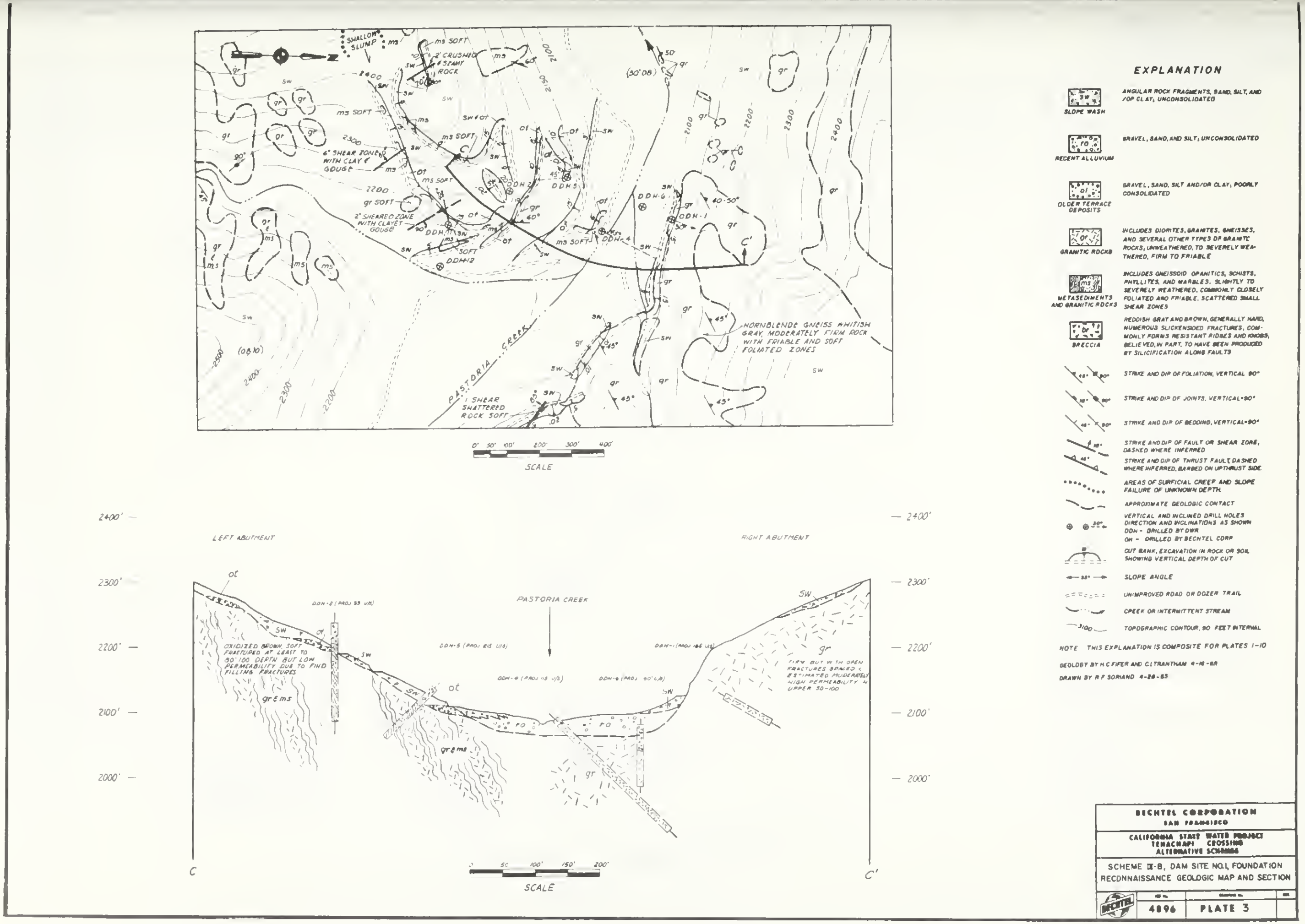
BECHTEL CORPORATION
SAN FRANCISCO

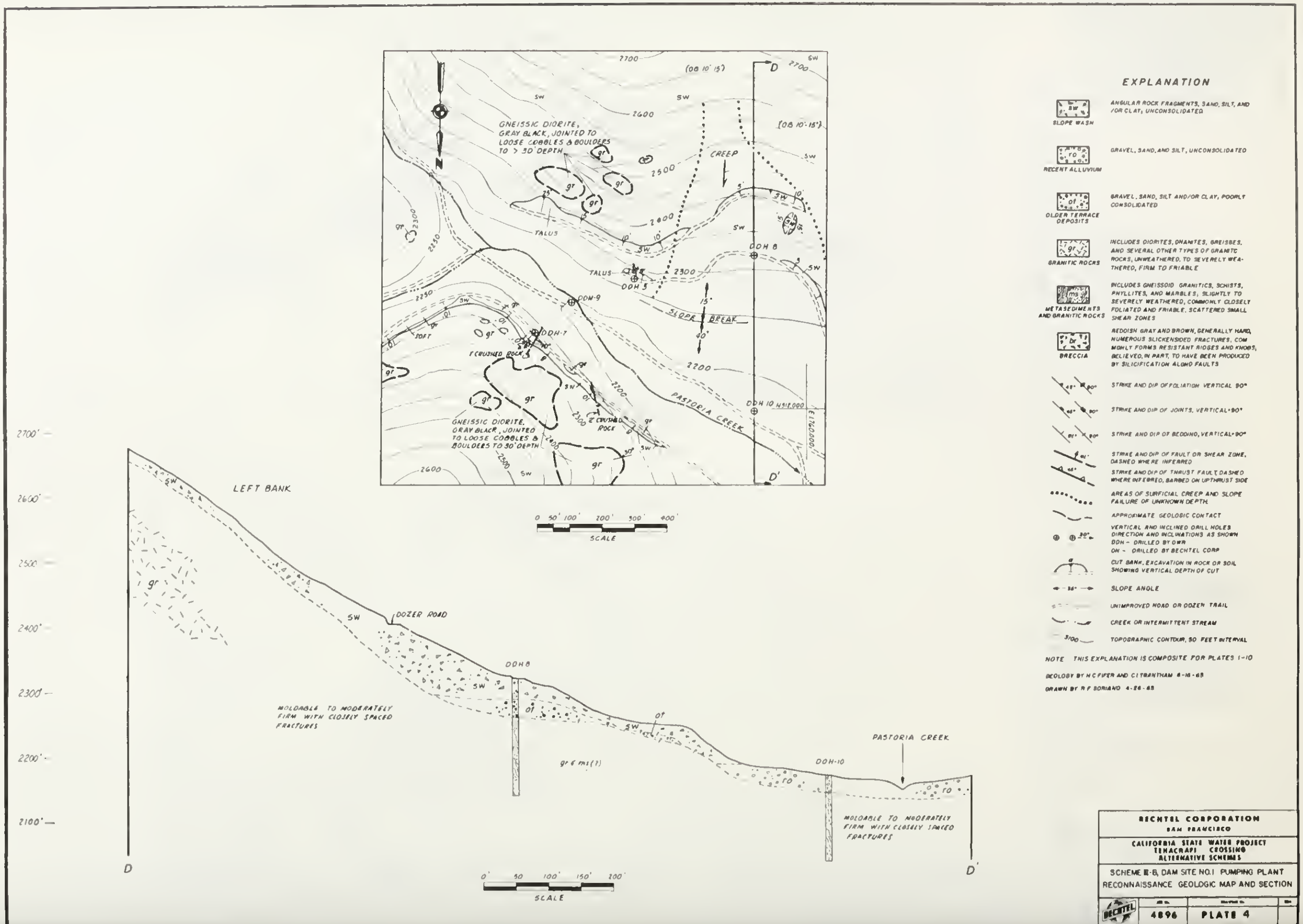
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES

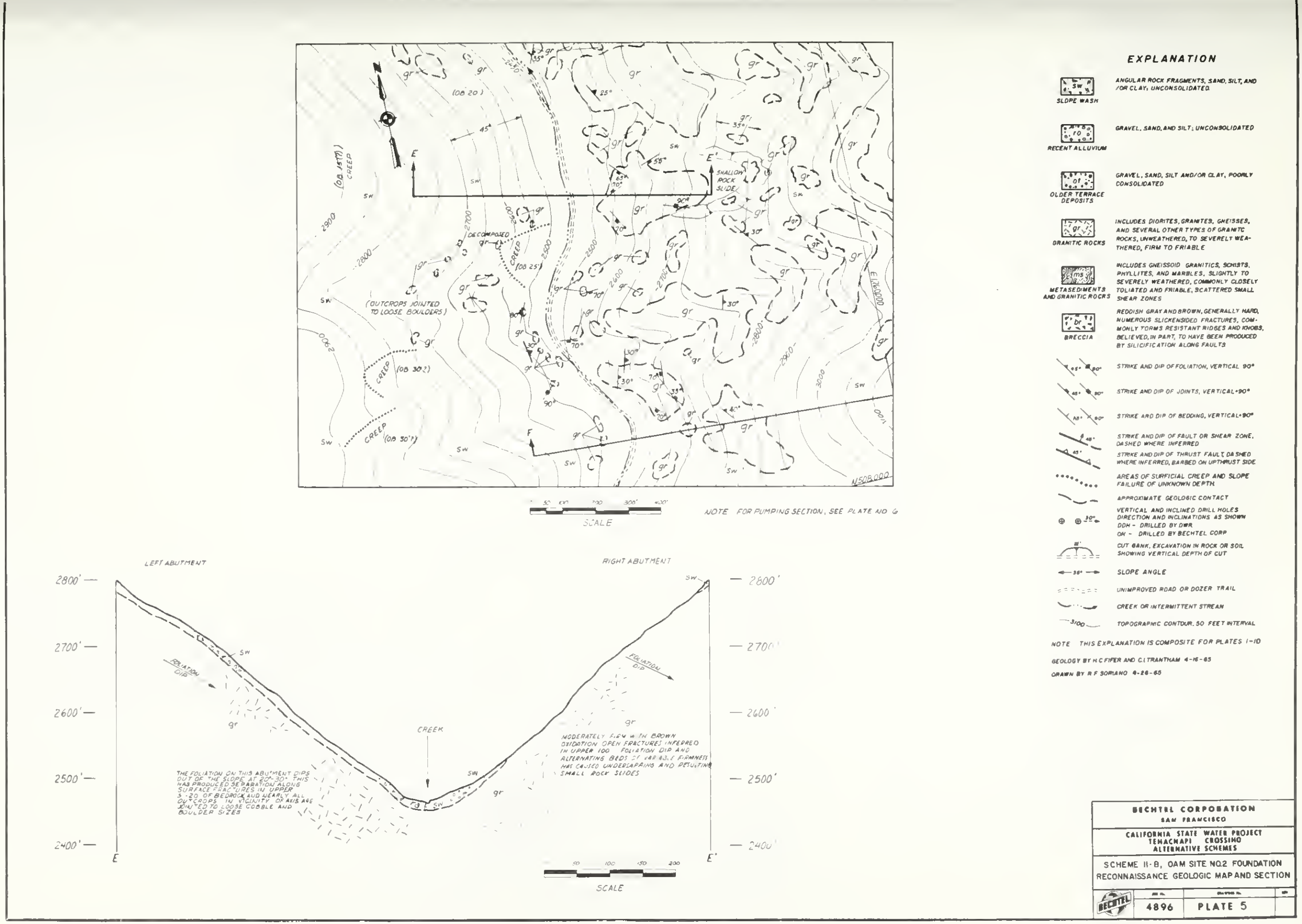
SCHEME II-B, DAM SITE NO1, FOUNDATION RECONNAISSANCE GEOLOGIC MAP AND SECTION

4896 PLATE I

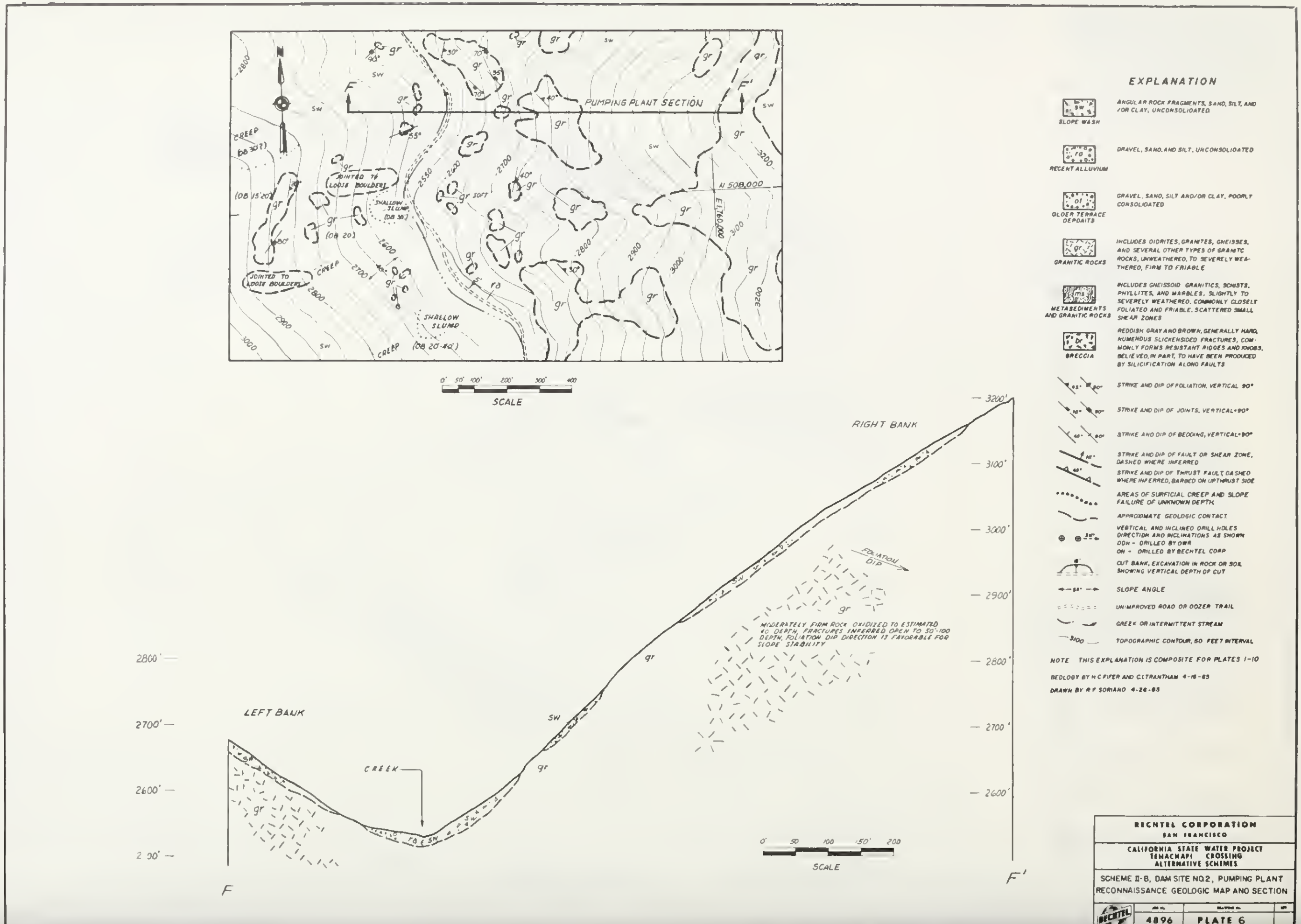

EXPLANATION
SLOPE WASH
ANGULAR ROCK FRAGMENTS, SAND, SILT, AND/OR CLAY, UNCONSOLIDATED
RECENT ALLUVIUM
GRAVEL, SAND, AND SILT; UNCONSOLIDATED
OLDER TERRACE DEPOSITS
GRAVEL, SAND, SILT AND/OR CLAY; POORLY CONSOLIDATED
GRANITIC ROCKS
INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS, UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS
INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES, SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE. SCATTERED SMALL SHEAR ZONES
BRECCIA
REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL 90°
STRIKE AND DIP OF BEDDING, VERTICAL 90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H C FIFER AND C I TRANTHAM 4-16-63
DRAWN BY R F SORIANO 4-26-63
TRANSMISSION LINES
PASTORIA CREEK
PARTLY JOINTED TO BOULDERS
HARD R'S STANDING 10 ABOVE VALLEY FLOOR
SOFT TO MODERATELY FIRM WEATHERED, BROWN
SCALE
LEFT BANK
FIRM
2600
2500'
2400'
2300'
2200'
2100'
2000'
1900'
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME II-A, DAM SITE NO1, PUMPING PLANT
RECONNAISSANCE GEOLOGIC MAP AND SECTION
4896
PLATE 2

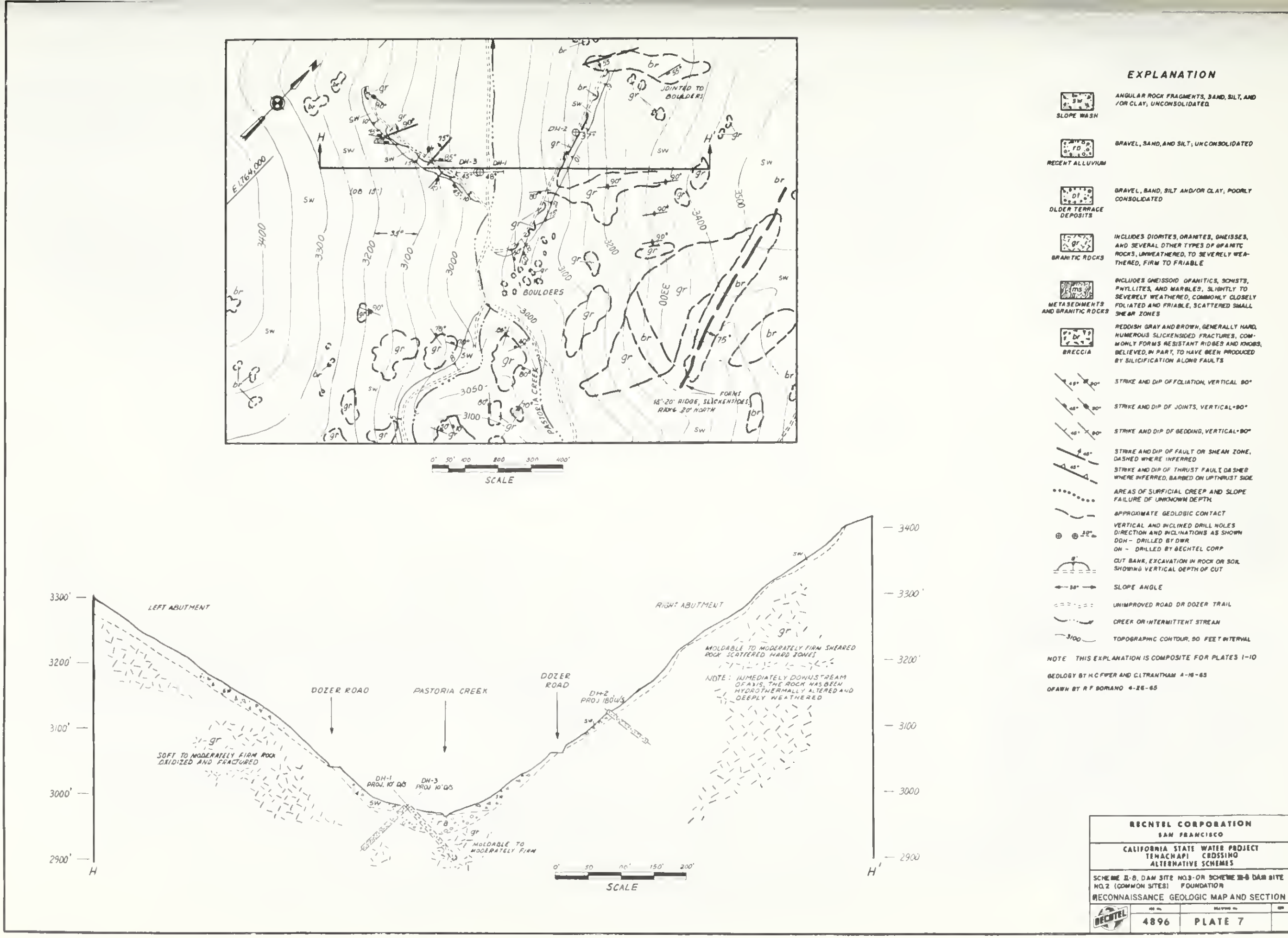

EXPLANATION
SLOPE WASH
ANGULAR ROCK FRAGMENTS, SAND, SILT, AND/OR CLAY; UNCONSOLIDATED
RECENT ALLUVIUM
GRAVEL, SAND, AND SILT; UNCONSOLIDATED
OLDER TERRACE DEPOSITS
GRAVEL, SAND, SILT AND/OR CLAY; POORLY CONSOLIDATED
GRANITIC ROCKS
INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS; UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS
INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES. SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE, SCATTERED SMALL SHEAR ZONES
BRECCIA
REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL 90°
STRIKE AND DIP OF BEDDING, VERTICAL 90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H C FIFER AND C.I TRANTHAM 4-16-68
DRAWN BY R F SORIANO 4-26-63
SHALLOW SLUMP
2' CRUSHED & SEAMY ROCK
6" SHEAR ZONE WITH CLAY & GOUGE
2" SHEARED ZONE WITH CLAYEY GOUGE
1' SHEAR SHATTERED ROCK SOFT
HORNBLENDE GNEISS WHITISH GRAY, MODERATELY FIRM ROCK WITH FRIABLE AND SOFT FOLIATED ZONES
PASTORIA CREEK
SCALE
LEFT ABUTMENT
RIGHT ABUTMENT
OXIDIZED BROWN, SOFT FRACTURED AT LEAST TO 80'-100' DEPTH BUT LOW PERMEABILITY DUE TO FIND FILLING FRACTURES
C
C'
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME II-B, DAM SITE NO.1, FOUNDATION RECONNAISSANCE GEOLOGIC MAP AND SECTION
4896
PLATE 3

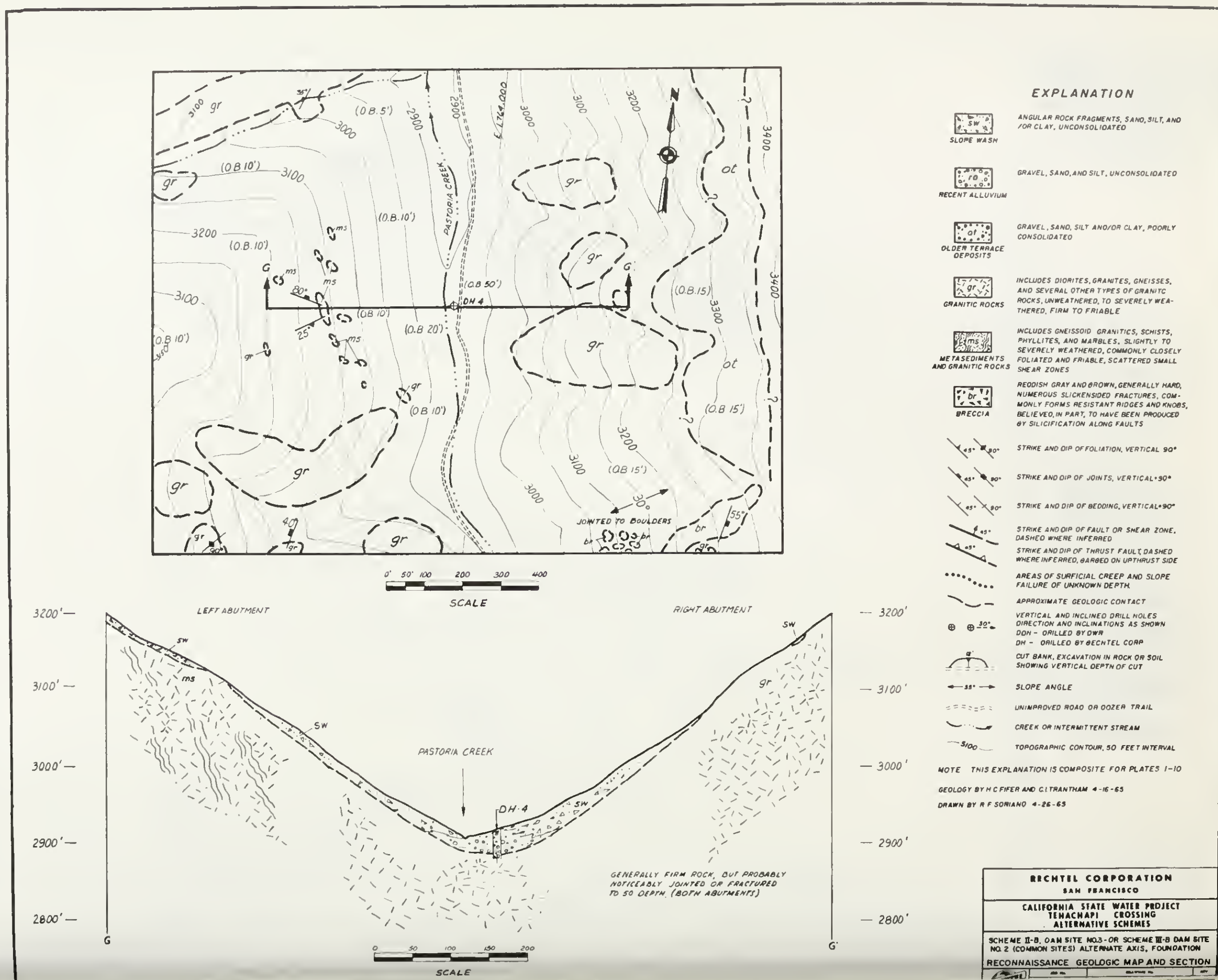

EXPLANATION
SLOPE WASH
ANGULAR ROCK FRAGMENTS, SAND, SILT, AND /OR CLAY; UNCONSOLIDATED
RECENT ALLUVIUM
GRAVEL, SAND, AND SILT, UNCONSOLIDATED
OLDER TERRACE DEPOSITS
GRAVEL, SAND, SILT AND/OR CLAY; POORLY CONSOLIDATED
GRANITIC ROCKS
INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS; UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS
INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES; SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE, SCATTERED SMALL SHEAR ZONES
BRECCIA
REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL 90°
STRIKE AND DIP OF BEDDING, VERTICAL 90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GNEISSIC DIORITE, GRAY BLACK, JOINTED TO LOOSE COBBLES & BOULDERS TO > 30' DEPTH
GNEISSIC DIORITE, GRAY BLACK, JOINTED TO LOOSE COBBLES & BOULDERS TO 30' DEPTH
CREEP
TALUS
SLOPE BREAK
PASTORIA CREEK
CRUSHED ROCK
SOFT
DOH 8
DOH 5
DOH-9
DOH-7
DOH 10
SCALE
LEFT BANK
DOZER ROAD
MOLDABLE TO MODERATELY FIRM WITH CLOSELY SPACED FRACTURES
gr & ms(?)
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME III-B, DAM SITE NO.1 PUMPING PLANT
RECONNAISSANCE GEOLOGIC MAP AND SECTION
4896
PLATE 4

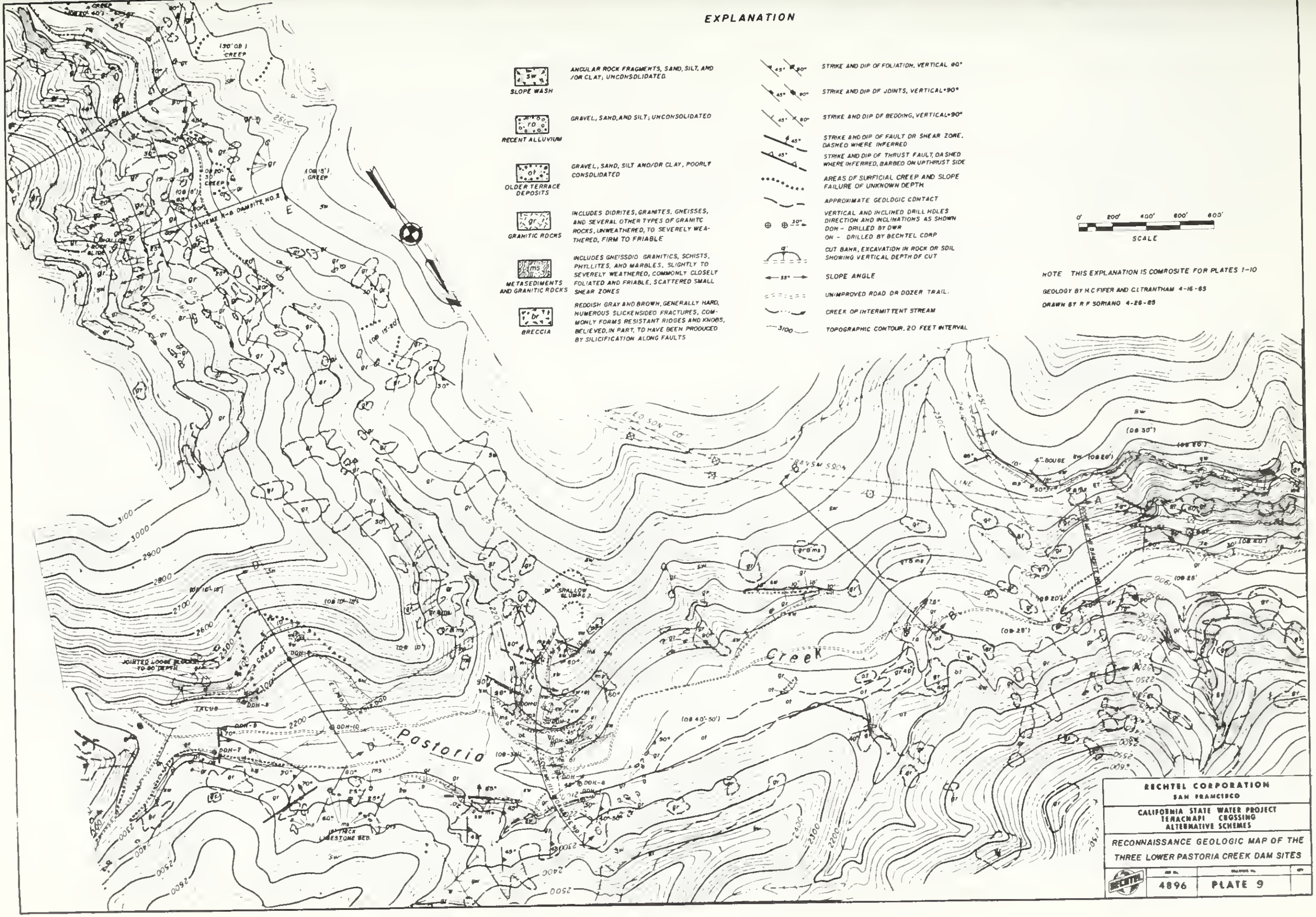


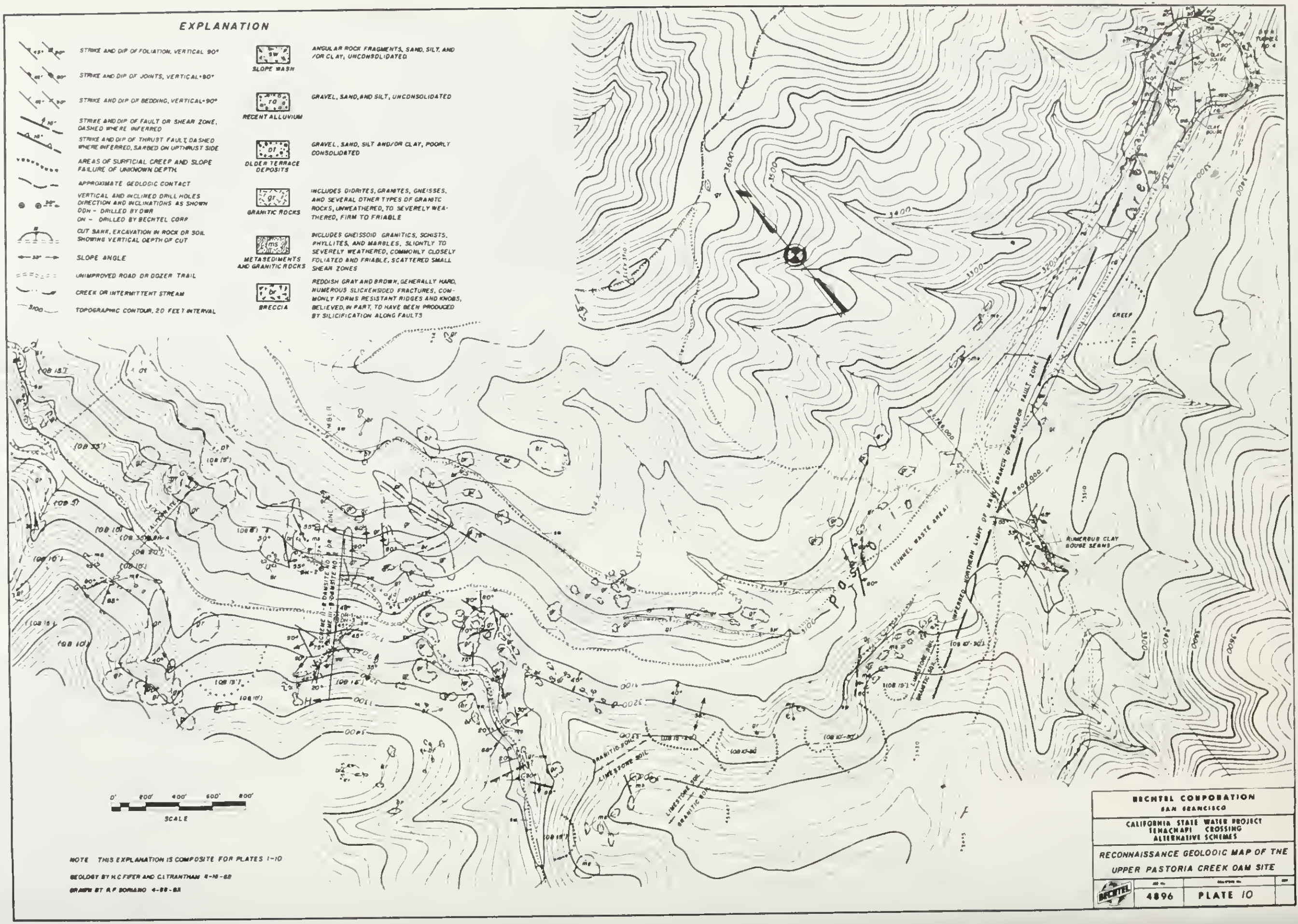
EXPLANATION
SLOPE WASH
ANGULAR ROCK FRAGMENTS, SAND, SILT, AND /OR CLAY; UNCONSOLIDATED
RECENT ALLUVIUM
GRAVEL, SAND, AND SILT; UNCONSOLIDATED
OLDER TERRACE DEPOSITS
GRAVEL, SAND, SILT AND/OR CLAY, POORLY CONSOLIDATED
GRANITIC ROCKS
INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITC ROCKS, UNWEATHERED, TO SEVERELT WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCRS
INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES, SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY TOLIATED AND FRIABLE, SCATTERED SMALL SHEAR ZONES
BRECCIA
REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY TORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL•90°
STRIKE AND DIP OF BEDDING, VERTICAL•90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H C FIFER AND C.I TRANTHAM 4-16-65
DRAWN BY R F SORIANO 4-26-65
SHALLOW ROCK SLIDE
DECOMPOSED
CREEP
(OUTCROPS JOINTED TO LOOSE BOULDERS)
E1,760,000
N508,000
SCALE
NOTE FOR PUMPING SECTION, SEE PLATE NO 6
LEFT ABUTMENT
RIGHT ABUTMENT
FOLIATION DIP
CREEK
THE FOLIATION ON THIS ABUTMENT DIPS OUT OF THE SLOPE AT 20°-30° THIS HAS PRODUCED SEPARATION ALONG SURFACE FRACTURES IN UPPER 5-20 OF BEDROCK AND NEARLY ALL OUTCROPS IN VICINITY OF AXIS ARE JOINTED TO LOOSE COBBLE AND BOULDER SIZES
MODERATELY FIRM WITH BROWN OXIDATION OPEN FRACTURES INFERRED IN UPPER 100 FOLIATION DIP AND ALTERNATING BEDS OF VARIABLE FIRMNESS HAS CAUSED UNDERSAPPING AND RESULTING SMALL ROCK SLIDES
2800'
2700'
2600'
2500'
2400'
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME II-B, OAM SITE NO.2 FOUNDATION RECONNAISSANCE GEOLOGIC MAP AND SECTION
4896
PLATE 5


PUMPING PLANT SECTION
F
F'
CREEP (DB 30-7)
(DB 15 20)
JOINTED TO LOOSE BOULDERS
SHALLOW SLUMP (DB 30)
SHALLOW SLUMP
CREEP (DB 20-40)
N 508,000
E 1,760,000
SCALE
RIGHT BANK
LEFT BANK
CREEK
FOLIATION DIP
MODERATELY FIRM ROCK OXIDIZED TO ESTIMATED 40' DEPTH, FRACTURES INFERRED OPEN TO 50'-100' DEPTH, FOLIATION DIP DIRECTION IS FAVORABLE FOR SLOPE STABILITY
EXPLANATION
SLOPE WASH — ANGULAR ROCK FRAGMENTS, SAND, SILT, AND/OR CLAY, UNCONSOLIDATED
RECENT ALLUVIUM — GRAVEL, SAND, AND SILT, UNCONSOLIDATED
OLDER TERRACE DEPOSITS — GRAVEL, SAND, SILT AND/OR CLAY, POORLY CONSOLIDATED
GRANITIC ROCKS — INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS, UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS — INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES, SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE. SCATTERED SMALL SHEAR ZONES
BRECCIA — REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL 90°
STRIKE AND DIP OF BEDDING, VERTICAL 90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN DDH - DRILLED BY DWR DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H C FIFER AND C L TRANTHAM 4-16-65
DRAWN BY R F SORIANO 4-26-65
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME II-B, DAM SITE NO.2, PUMPING PLANT
RECONNAISSANCE GEOLOGIC MAP AND SECTION
4896
PLATE 6


EXPLANATION
SLOPE WASH
ANGULAR ROCK FRAGMENTS, SAND, SILT, AND/OR CLAY; UNCONSOLIDATED
RECENT ALLUVIUM
GRAVEL, SAND, AND SILT; UNCONSOLIDATED
OLDER TERRACE DEPOSITS
GRAVEL, SAND, SILT AND/OR CLAY; POORLY CONSOLIDATED
GRANITIC ROCKS
INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS, UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS
INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES, SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE, SCATTERED SMALL SHEAR ZONES
BRECCIA
REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL-90°
STRIKE AND DIP OF BEDDING, VERTICAL-90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H.C FIFER AND C.L TRANTHAM 4-16-65
DRAWN BY R F ROMANO 4-26-65
BOULDERS
JOINTED TO BOULDERS
PASTORIA CREEK
FORMS 15'-20' RIDGE, SLICKENSIDES, RAKE 20° NORTH
E 1,764,000
DH-1
DH-2
DH-3
SCALE
LEFT ABUTMENT
RIGHT ABUTMENT
DOZER ROAD
PASTORIA CREEK
DOZER ROAD
DH-1 PROJ. 10' D/S
DH-3 PROJ. 10' D/S
DH-2 PROJ. 180' U/S
SOFT TO MODERATELY FIRM ROCK OXIDIZED AND FRACTURED
MOLDABLE TO MODERATELY FIRM
MOLDABLE TO MODERATELY FIRM SHEARED ROCK SCATTERED HARD ZONES
NOTE: IMMEDIATELY DOWNSTREAM OF AXIS, THE ROCK HAS BEEN HYDROTHERMALLY ALTERED AND DEEPLY WEATHERED
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME II-8, DAM SITE NO.3 - OR SCHEME III-8 DAM SITE NO.2 (COMMON SITES) FOUNDATION
RECONNAISSANCE GEOLOGIC MAP AND SECTION
4896
PLATE 7


3400
3300
3200
3100
3000
2900
ot
gr
ms
br
(0.B.15')
(0.B 50')
(0.B 20')
(0.B 10')
(0.B 5')
DH 4
G
G'
PASTORIA CREEK
JOINTED TO BOULDERS
0' 50' 100 200 300 400
SCALE
LEFT ABUTMENT
RIGHT ABUTMENT
PASTORIA CREEK
DH-4
SW
3200'
3100'
3000'
2900'
2800'
GENERALLY FIRM ROCK, BUT PROBABLY NOTICEABLY JOINTED OR FRACTURED TO 50 DEPTH. (BOTH ABUTMENTS)
0 50 100 150 200
SCALE
EXPLANATION
SLOPE WASH
ANGULAR ROCK FRAGMENTS, SAND, SILT, AND /OR CLAY, UNCONSOLIDATED
RECENT ALLUVIUM
GRAVEL, SAND, AND SILT, UNCONSOLIDATED
OLDER TERRACE DEPOSITS
GRAVEL, SAND, SILT AND/OR CLAY, POORLY CONSOLIDATED
GRANITIC ROCKS
INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS, UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS
INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES, SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE, SCATTERED SMALL SHEAR ZONES
BRECCIA
REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL 90°
STRIKE AND DIP OF BEDDING, VERTICAL 90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 50 FEET INTERVAL
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H C FIFER AND C I TRANTHAM 4-16-65
DRAWN BY R F SORIANO 4-26-65
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
SCHEME II-B, DAM SITE NO.3-OR SCHEME III-B DAM SITE NO. 2 (COMMON SITES) ALTERNATE AXIS, FOUNDATION
RECONNAISSANCE GEOLOGIC MAP AND SECTION


EXPLANATION
SLOPE WASH — ANGULAR ROCK FRAGMENTS, SAND, SILT, AND/OR CLAY; UNCONSOLIDATED
RECENT ALLUVIUM — GRAVEL, SAND, AND SILT; UNCONSOLIDATED
OLDER TERRACE DEPOSITS — GRAVEL, SAND, SILT AND/OR CLAY, POORLY CONSOLIDATED
GRANITIC ROCKS — INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS, UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS — INCLUDES GNEISSOID GRANITICS, SCHISTS, PHYLLITES, AND MARBLES, SLIGHTLY TO SEVERELY WEATHERED, COMMONLY CLOSELY FOLIATED AND FRIABLE, SCATTERED SMALL SHEAR ZONES
BRECCIA — REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL 90°
STRIKE AND DIP OF BEDDING, VERTICAL 90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 20 FEET INTERVAL
0' 200' 400' 600' 800'
SCALE
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H C FIFER AND C L TRANTHAM 4-16-85
DRAWN BY R F SORIANO 4-26-85
Pastoria
Creek
SCHEME II-B DAMSITE NO. 1
SCHEME III DAMSITE NO. 2
SCHEME II-B DAMSITE NO. 3
SHALLOW SLUMPS
JOINTED LOOSE BLOCKS TO 80' DEPTH
TALUS
LIMESTONE BED
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
RECONNAISSANCE GEOLOGIC MAP OF THE THREE LOWER PASTORIA CREEK DAM SITES
4896
PLATE 9


EXPLANATION
STRIKE AND DIP OF FOLIATION, VERTICAL 90°
STRIKE AND DIP OF JOINTS, VERTICAL-90°
STRIKE AND DIP OF BEDDING, VERTICAL-90°
STRIKE AND DIP OF FAULT OR SHEAR ZONE, DASHED WHERE INFERRED
STRIKE AND DIP OF THRUST FAULT, DASHED WHERE INFERRED, BARBED ON UPTHRUST SIDE
AREAS OF SURFICIAL CREEP AND SLOPE FAILURE OF UNKNOWN DEPTH
APPROXIMATE GEOLOGIC CONTACT
VERTICAL AND INCLINED DRILL HOLES DIRECTION AND INCLINATIONS AS SHOWN
DDH - DRILLED BY DWR
DH - DRILLED BY BECHTEL CORP
CUT BANK, EXCAVATION IN ROCK OR SOIL SHOWING VERTICAL DEPTH OF CUT
SLOPE ANGLE
UNIMPROVED ROAD OR DOZER TRAIL
CREEK OR INTERMITTENT STREAM
TOPOGRAPHIC CONTOUR, 20 FEET INTERVAL
SLOPE WASH
ANGULAR ROCK FRAGMENTS, SAND, SILT, AND /OR CLAY, UNCONSOLIDATED
RECENT ALLUVIUM
GRAVEL, SAND, AND SILT, UNCONSOLIDATED
OLDER TERRACE DEPOSITS
GRAVEL, SAND, SILT AND/OR CLAY, POORLY CONSOLIDATED
GRANITIC ROCKS
INCLUDES DIORITES, GRANITES, GNEISSES, AND SEVERAL OTHER TYPES OF GRANITIC ROCKS, UNWEATHERED, TO SEVERELY WEATHERED, FIRM TO FRIABLE
METASEDIMENTS AND GRANITIC ROCKS
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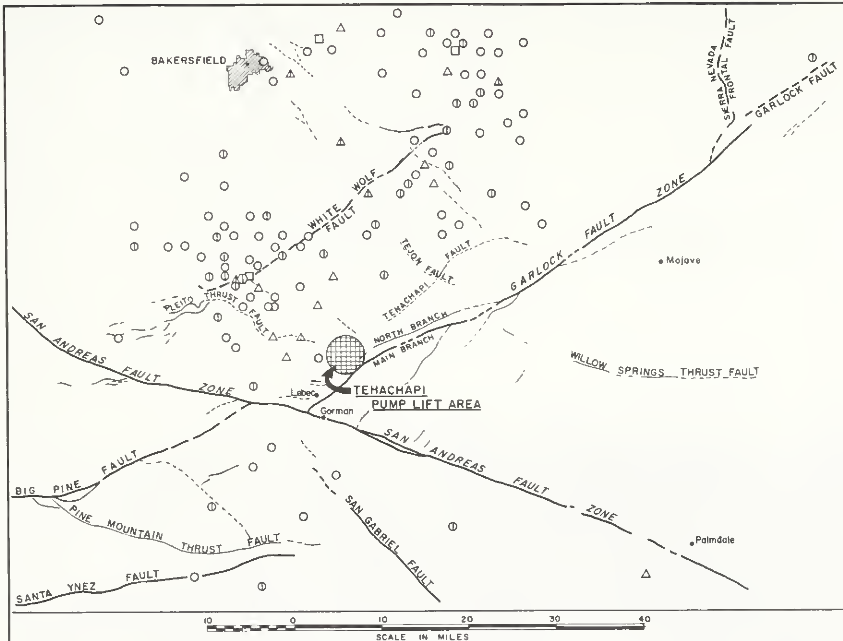
BRECCIA
REDDISH GRAY AND BROWN, GENERALLY HARD, NUMEROUS SLICKENSIDED FRACTURES, COMMONLY FORMS RESISTANT RIDGES AND KNOBS, BELIEVED, IN PART, TO HAVE BEEN PRODUCED BY SILICIFICATION ALONG FAULTS
0' 200' 400' 600' 800'
SCALE
NOTE THIS EXPLANATION IS COMPOSITE FOR PLATES 1-10
GEOLOGY BY H.C FIFER AND C.L TRANTHAM
CREEP
NUMEROUS CLAY GOUGE SEAMS
(TUNNEL WASTE AREA)
INFERRED NORTHERN LIMIT OF MAIN BRANCH OF GARLOCK FAULT ZONE
LIMESTONE SOIL
GRANITIC SOIL
BECHTEL CORPORATION
SAN FRANCISCO
CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES
RECONNAISSANCE GEOLOGIC MAP OF THE UPPER PASTORIA CREEK DAM SITE
4896
PLATE 10



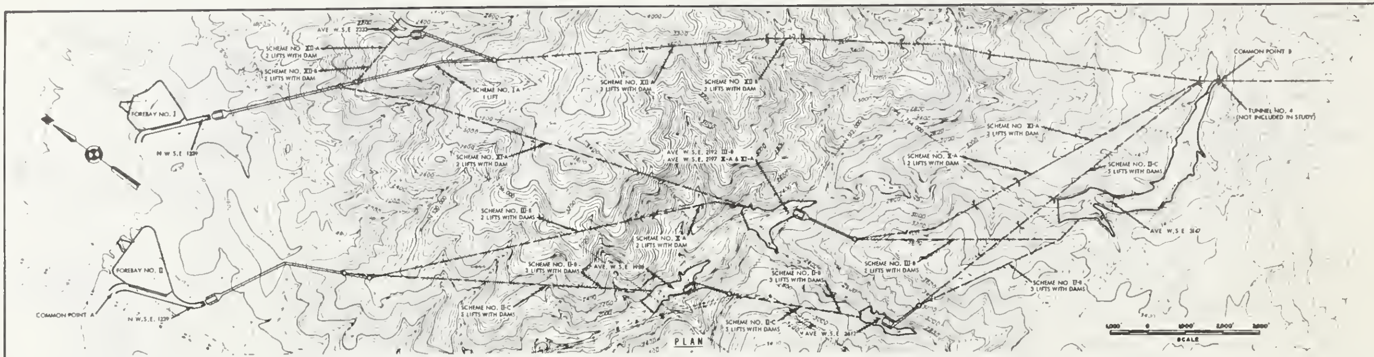
THE APPROXIMATE LOCATIONS OF EARTHQUAKE EPICENTERS WHICH HAVE OCCURRED IN THE VICINITY OF THE TEHACHAPI PUMPLIFT AREA BETWEEN THE YEARS 1934 AND 1960.

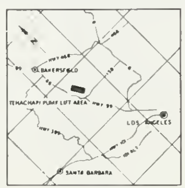
INFORMATION FROM:
"EARTHQUAKE EPICENTER AND FAULT MAP OF CALIFORNIA",
COMPILED BY D.M. HILL, V.A. MOORE, J.E WOLFE, AND C.A. LAO.

SYMBOL	MAGNITUDE
○	4.0 - 4.4
⦶	4.5 - 4.9
△	5.0 - 5.4
△⃒	5.5 - 5.9
□	6.0 - 6.4
⦶	7.5 - 7.9

NOTE:
ALL EPICENTERS SHOWN HAVE BEEN INSTRUMENTALLY DETERMINED AND ARE PRESENTED ACCORDING TO THE RICHTER SCALE OF MAGNITUDE.

TEHACHAPI PUMP LIFT
EARTHQUAKE EPICENTER MAP
PLATE II





LOCATION MAP

LEGEND

PUMPING PLANT
DISCHARGE PIPES
SURGE TANK
TUNNEL
SIPHON
DAM AND RESERVOIR

NOTES

TOPOGRAPHY OBTAINED FROM MAP COMPILED BY STEREOPHOTOGRAMMETRIC METHODS FROM PHOTOGRAPHY DATED JULY 21, 1956

GRID BASED ON CALIFORNIA STATE PLANE COORDINATE SYSTEM, ZONE 5.

BECHTEL CORPORATION
SAN FRANCISCO

CALIFORNIA STATE WATER PROJECT
TEHACHAPI CROSSING
ALTERNATIVE SCHEMES

LOCATION MAP
AND COMPARISON OF PLANS

4896 | PLATE 12

TEHACHAPI PUMP LIFT SYSTEM

GENERAL REMARKS

by
Mr. Julian Hinds, Consultant to Bechtel
May, 1965

Mr. Chairman and Members of the Tehachapi Crossing Consulting Board, I wish to express to you my appreciation for the privilege of appearing before you in today's deliberations. My appearance at this particular point, near the end of a summary of months of work and study, perhaps carries with it a responsibility beyond my ability to bear. However, I have had a life-time of experience in the water works field, and would like to say a few words which I hope may be helpful.

It is trite to tell you that the Tehachapi Crossing poses a great problem, one of many on the world's greatest domestic water supply project. Most of us here today have spent months accumulating facts concerning it. The time now has come to lay these facts on the table, end to end, and examine them candidly, entirely free from sentimentality. My previous associations with each of the organizations involved, and with many of their employees and advisors, leave no room to doubt that this can and will be done. I am confident that this is the attitude of all of us here today.

I shall not go into the technicalities of pump design or testing. Others present are better prepared to do this. What I have to say is non-technical - it will concern only a few guiding principles, or policies, in arriving at the major decisions that must be made.

First of all, this complex problem involves many people. People are human and as such sometimes get fixed or preconceived ideas. I have no indication whatever that such ideas have entered into the Tehachapi study. But to guard against their possible hidden existence, or their subsequent development, permit me to caution against the even small chance that human frailty might creep in. To guard against this, we must very carefully and objectively examine all of the ideas that have been advanced and make sure that they are evaluated without any semblence of preconceived preference.

My second non-technical point concerns the time element - the urgent need to get the project under construction. This is sometimes thought of as the State's problem, but actually it is particularly pertinent to Southern California. If the future can be judged by the past, water cannot flow over the Tehachapis too soon. Time certainly is of the essence but, within reason, it is only one of many essentials. The date of need cannot be chosen to a day or a month, and is capable of reasonable adjustment to other factors. It is recognized that in all important endeavors there comes a time when discussion must cease and action must begin. But, in my opinion, the penalty of setting that time too soon could be greater than the risk of some delay.

Now, bringing together the two guiding principles which I have proposed and applying them to the problem at hand, I suggest that they point to the following decisions at this time:

1. We should proceed with great caution before adopting a single-lift system with its most complex problems of high-head penstocks and unproven pumps. This solution I feel would be subject to the most possibility of errors of human frailty.

2. At the opposite extreme, quite understandably, it would be human nature to settle upon the entirely conventional solution of a three-lift system, since this would involve the least risks in extending our horizons. For this reason I suggest that we also should examine the three-lift solution with great caution before adopting this "easy way out". This project is too important to rely only upon the conventional.

3. There remains, then, a two-lift solution which, in my judgement, offers the most reasonable compromise between pushing out too far or not far enough. It is in connection with this recommended solution that I have suggested the second guiding principle - that of not squeezing the time element too short. A two-lift solution would permit consideration of at least two and possibly three alternative types of pumps. The best pump can be selected only after completion of a thorough program of testing and other research. I believe that a reasonable allowance of time, disregarding any presently scheduled date, would be a good investment.

Julian Hinds

Julian Hinds
Consulting Engineer

May 3, 1965
JH/ds

TEHACHAPI PUMP LIFT

Presentation to
California Department of Water Resources
and the
Tehachapi Crossing Consulting Board

by

R. A. Skinner
General Manager and Chief Engineer
The Metropolitan Water District of Southern California

May 5, 1965

A principal objective of this statement is to summarize and evaluate the evidence which has come to the attention of the Metropolitan Water District regarding alternative pumping systems and types of pumping equipment that have been given consideration for the Tehachapi lift. Sources of information considered include reports by the Department of Water Resources and its consultants and by Metropolitan's consultants, as well as investigations by Metropolitan's personnel and their consultations with informed technologists.

It is observed first that the problem of selecting the optimum pumping system and type of pumping units does not yield readily or with certainty to an unique solution, particularly considering the necessity now for a prompt decision on the primary issue. Comparative superiority of certain alternative systems and types of units in respect to important characteristics can be identified with assurance from accumulated evidence; in other cases there may be considerable evidence to support a presumption of superiority, without the available information being sufficiently conclusive to be classified as proof.

In considering the possibility of optimization, questions of comparative advantages of different types of pumping equipment, motors, penstocks, valves, and related facilities must be evaluated in conjunction with environmental factors of topography, geology, slope stability, stream flow and debris transport, location and construction of forebays, and other factors peculiar to the available sites.

Because of the District's favorable experience with single-stage pumps on its Colorado River Aqueduct, a request was made during the earlier stages of the investigations of the Tehachapi crossing that the Department study more extensively the use of three lifts for the pumping system. This would permit single-stage pumps to be used. The District contended that single-stage pumps of American design were the most efficient pumps that could be used for the Tehachapi crossing and therefore should be given full consideration. Further, the District suggested that complete and exhaustive tests of models of the several types of pumps considered competitively applicable should be made in one adequately equipped laboratory. Also, the relatively severe problems associated with motors, valves, and delivery lines for a single-lift system were discussed.

There is substantial evidence both from model testing and prototype performance to indicate that, within applicable limitation as to head, the single-stage, single-suction volute centrifugal pump is the most efficient, most reliable, and most easily maintained device for lifting water at the comparatively high rates of flow desirable for the Tehachapi lift pumping units. Among the reasons for this are the favorable inlet conditions, with no shaft through the eye of the impeller, and the advantage of the single impeller and volute in minimizing skin friction and turbulence. These advantages would militate toward adoption of a three-lift system if the environmental factors are as favorable therefor as for a two-lift or single-lift system. An additional advantage of a three-lift system is the opportunity for compatibility with favorable electric drive motor characteristics in regard to size and speed of units. The three-lift system also brings the plate thickness and type of steel for the penstocks well within the scope of reliable quality control.

The estimated total capital and annual costs of a number of alternative systems for the Tehachapi crossing have been investigated by Bechtel Corporation. One of the most favorable alternative systems, based on the present worth of total costs, is a three-lift system along

Pastoria Creek using single-stage, single-suction vertical pumps. If the environmental factors of geology, slope stability, and potential effects of floods can be adequately provided for in the engineering design, the District would consider such a three-lift system on the Pastoria Creek alinement the most advantageous of all systems.

Information available to the District at the present time is not sufficiently complete for adequate evaluation of the comparative effects of geological conditions and other environmental factors on alternative systems and locations.

The three-lift system concept with the necessary two additional forebays, two additional pumping stations, and correspondingly greater number of pumping units and accessories as compared with a single-lift system has been given a severely reduced reliability rating in comparative analyses presented by the Department and its consultants. A corresponding down-grading of reliability rating has been accorded the two-lift system concept in comparison with a single-lift system. Aside from reliability as a function of environmental factors, it is necessary to evaluate the comparative reliability of different types of units in alternative systems containing different numbers of units. If a numerical rating system is applied, the resulting comparative values include the effects on ratings of both the characteristics of pumping units, and the number of units in series. Methods of weighting these different factors can critically affect the relative numerical ratings derived.

On the Pastoria Creek location, a reservoir at each plant would provide flexibility in starting and stopping the pumping units. A reservoir at the upper end of the third lift would provide desirable storage at the top of the pump lift. The operation of three plants in series would be no more difficult than the operation of the Buena Vista, Wheeler Ridge No. 1, and Wheeler Ridge No. 2 pumping plants immediately ahead of the Tehachapi pumping system. In fact, with a uniform flow of water through the Tehachapi system the operating problems for three lifts would be less complicated than for the three plants upstream where there may be a different rate of flow of water through each of the plants at any time because of diversions from the reaches between plants.

The two-lift alternative represents an intermediate concept of considerable persuasiveness. Two-stage, double-suction pumps in the head-capacity range representative of this alternative are in use in a number of existing installations with exceptionally good results in regard to both efficiency and reliability. Model tests as well as observation of prototype units in operation indicate relatively high efficiency for this type of pump. Two-stage, double-suction pumps at Limberg, Erzhausen, Ffestiniog, Vianden, and Provvidenza are operating successfully at both head and capacity approximating or exceeding the requirements at Tehachapi for a two-lift system.

For pumping units with horizontal shafts the double-suction pumps provide the additional advantage of hydraulic symmetry with virtual elimination of load on thrust bearings. By the use of split casings there can be obtained the additional advantage of easy access, the horizontal setting enabling the crane to be spotted over any component of the assembly without need to remove other parts, and positioning of impellers and seals can be accurately controlled. With the horizontal setting the centers of gravity of the heavy parts of the machines are at minimum distance from anchorage at the machine hall floor, which is a favorable relationship for resistance to seismic accelerations.

The record of performance of two-stage, double-suction pumps with horizontal shafts is excellent in the European plants where such units are in service. The report of Daniel, Mann, Johnson, and Mendenhall on Investigation of High Head Pumping Practice in Europe, dated October 1964, pointed out that such units had better operating records than did the vertical units of this type. Professor Gerber and Dr. Sutherland in their report for Bechtel on European practices state that wherever split housings were provided, the management of the plants endorsed strongly their advantages for facilitating overhaul.

The results of tests of the two-stage, double-suction pump model at the J. M. Voith plant were gratifying. Efficiency was better than expected, although model tests indicate that it is about 2 per cent below the efficiency that can be obtained by the use of single-stage pumps on a three-lift system.

A distinct advantage of the two-stage, double-suction pump is that the problem of wear in the seal rings from silt or sand in the water is greatly reduced because it is only necessary to seal the pressure in the second stage from the pressure developed by the first stage of the pump rather than against atmospheric pressure. Thus, the maximum pressure to be sealed is that developed by one-stage, or approximately 500 feet, as compared to 650 feet for a single-stage pump in a three-lift system and 2,000 feet for a four-stage pump in a single-lift system.

Since it appears that a suitable reservoir site has been located on the ridge location for storage at the second pumping plant of a two-lift system, the flexibility of operation for such an alternative would approximate that obtainable by the use of the Pastoria Creek alinement. The maximum pressure to be designed for on a two-lift system is on the order of 1,000 feet, which is appreciably higher than the pressure of 650 feet on a three-lift system but is only half that of the 2,000 feet on a single-lift system.

The experience with two-stage, double-suction pumps is extensive and, as Gerber and Sutherland point out in their report, all the characteristics of such pumps for a two-lift system at Tehachapi are well within the envelope of actual design and operation experience.

There is substantial evidence indicating that single-stage pumps would be suitable for a two-lift system, with head of about 980 feet for each lift, although existing installations do not provide sufficient observation of performance in the head-capacity magnitudes involved for conclusive demonstration of such suitability. Additional model testing may throw more light on this question, and several large units about to go into service may afford further observation. If a decision were made to adopt a two-lift system, selection of the type of pump could be deferred for about six months to permit additional time for comparative evaluation of pumping unit characteristics.

While there is no known example of the use of single-purpose, single-stage pumps for delivering water under the conditions of head, capacity, and continuous service operation that would be imposed for a two-lift system at Tehachapi, consultants for Bechtel Corporation have pointed out that Francis turbines which are similar in construction to a single-stage pump of the type required for Tehachapi have been operating against heads of a thousand feet and higher and have given satisfactory performance. In addition they point out that pumped storage projects have been built wherein reversible pump turbines operate successfully against comparable heads. Further, Bechtel Corporation has under way a testing program to determine the suitability of single-stage pumps for operation against heads up to 1,000 feet. However, at the present time it must be recognized that the use of single-stage pumps to operate against a head of 980 feet for the Tehachapi lift would be a major extrapolation beyond present experience. All evidence seems to indicate that such a pump will be completely developed in the near future and will be found the best pump for heads up to 1,000 feet. Nevertheless, it could not now be considered that use of a single-stage pump is supported by a history of reliable and efficient performance under such conditions.

DMJM is conducting tests on wear of metals by the passage of water at high velocity across a sample disc. These tests are of importance in the selection of materials for, and in the design of, the Tehachapi pumps. Caution is urged, however, in the interpretation of the results. Two pressure drops across identical samples are used, one at 500 feet and one at 1,000 feet. The wear with a 1,000-foot drop was much greater than with 500 feet, but it should not be concluded that the wear in properly designed seals on a pump operating against a 1,000-foot head necessarily will be greater than the wear in the seals on a pump operating at 500-foot head. If this were so the fourth stage outer seals on a four-stage pump operating against a head of 2,000 feet would be correspondingly more vulnerable. It is apparent that the seals must be designed for the applicable head in order to give reliable service. For the higher head, a longer seal or more tortuous labyrinth would be designed to control the velocity of water discharged through the seal, and mitigate the wear resulting from sediments.

Mechanical problems, such as designing the most reliable and effective seals, have not been fully explored and the adequate experience records of prototype installations are not available on very high head single-stage pumps such as would be required for a two-lift system.

Four-stage pumps have been proposed for the single-lift alternative. In the research and development program it has been postulated that these pumps each would have a capacity of 313 cfs and a speed of 600 rpm, resulting in a specific speed (GPM-feet basis) of 2160 for a total head of 1951 feet. Considering that the total rated delivery capability of the pumping system is now indicated to be 4,100 cfs instead of 5,000 cfs as previously assumed for purposes of the research and development, the pump capacity could be reduced if the same number of units (16 as previously assumed) were installed.

In contrast with the considerable number of existing installations of two-stage, double-suction pumps with head and capacity fully comparable to the requirements for a two-lift system at Tehachapi, there is no existing installation of multistage pumps which approaches closely to the combination of head, capacity, and specific speed proposed for the pumps for a single-lift system at Tehachapi. The existing installations which have been referred to for supporting evidence and the comparable characteristics of the pumps are as follows:

Multistage Pump Installations

Installation		No. of Stages	Head (ft) Total	Head (ft) Per Stage	Capacity (cfs)	Speed of Rotation (rpm)	Specific Speed (gpm-ft)
Tehachapi (Proposed single-lift)		4	1951	483	313	600	2160
Lunersee	V-S-EW	5	3151	630	144	750	1500
Motec	S	3	2065	688	115	750	1270
Ponale	R	5	1903	381	130	500	1130
Etzel	S	5	1600	320	99	500	1400
Tierfehd	S	3	1755	585	97	1000	1750
Tremorgio	EW	9	2953	328	16.5	1000	1120

Manufacturers: EW: Escher Wyss R: Riva S: Sulzer V: Voith

While three of the multistage pumps represented in the foregoing tabulation are designed for greater head per stage than that of the four-stage pumps proposed for the single-lift alternative at Tehachapi, the capacities of the existing pumps are much less, ranging from 30 to 50 per cent of that proposed for the Tehachapi four-stage pump (disregarding the Tremorgio pumps, which have only 16.5 cfs capacity). The specific speed of the existing multistage pumps ranges from 1120 to 1750 as compared with 2160 indicated for the Tehachapi pumps, the latter presumptively having been proposed to be higher in an effort to improve efficiency and economy.

It has been indicated by pump technologists that appropriate design for the proposed higher-capacity Tehachapi four-stage pumps can be accomplished satisfactorily. Without questioning the possibility of adequate design, several related factors can be pointed out that are of significance in this regard. In the type of pump involved, water enters the single suction eye of the impeller through an annular space around the shaft of the pump. The shaft diameter must be large enough to insure that the first critical speed will be sufficiently above the runaway speed of the pump (in case of reversal of flow) to avoid injurious vibration. Increasing the diameter of the impeller, and consequently of the pump casing, to improve the hydraulic conditions without changing the output, would necessitate slowing down the speed of rotation and correspondingly reducing the specific speed, because the peripheral speed of the impeller is fixed unalterably by the head per stage. However, diminishing the specific speed presumptively has the effect of impairing efficiency, although it may be doubted that the optimum specific speed for maximizing efficiency for a four-stage pump of the head and capacity required for a single lift at Tehachapi has been determined, and even more seriously doubted whether the optimum specific speed for maximizing efficiency would be optimum also for maximizing reliability for continuous operation. In this connection, it may be significant that the specific speed of the pumps in the existing installations listed above, as well as in three other pumped storage plants under construction in Italy in which four-stage pumps are being installed, and one in which six-stage pumps are being installed, in every case is considerably below that proposed for the four-stage pumps for Tehachapi, ranging from 52 to 81 per cent of the specific speed proposed for the latter.

As increasing the shaft diameter and consequently the impeller and pump casing diameters unavoidably results both in increasing the cost and in diminishing the specific speed (and therefore possibly reducing the efficiency) the competitive aspects of design will tend to limit these diameters to the lowest magnitudes considered feasible. It is of interest that the four-stage model now under test under the R and D program has been below expectation as to efficiency. This development appears to have given rise to an opinion that the less favorable characteristics than expected somehow must be the result of a fortuitous deficiency in quality of the particular model rather than the characteristics normally to be expected from a four-stage pump model of the selected head-capacity-speed relationships. A more comprehensive R and D program than so far in prospect would be necessary to achieve dependable answers to the very critical problems involved, not disregarding the aspects of cavitation and requirements for submergence.

Tests on the model of the four-stage pump for a single-lift system indicate that the efficiency of the prototype would be substantially below that of an alternative two-stage, double-suction pump for a two-lift system, and even more below that of a single-stage pump for a three-lift system. It is necessary at the fourth stage of the pump to seal a pressure of 2,000 feet within the volute or to the atmosphere on one side of the impeller. On the other side, the seal is exposed only to the differential pressure between that in the volute and that at the discharge of the third stage. The high-pressure seal has been maintained on some of the pumps in Europe without difficulty; however, the presence of sediment in the water will require special attention in the design of this high-pressure labyrinth seal on the balancing plate.

A single-lift system with the high pressures in the valves and delivery lines was found by Bechtel to have the highest present worth of total cost. The complexity and lower efficiency of the pumping unit, the high-pressure seal, the longer time required for repair of the unit, and the higher penstock pressure are additional disadvantages of the single-lift system. On the other hand, a single pumping plant would have an advantage of somewhat simpler operation, although this is the sixth pumping plant in the series and must be operated in coordination with other plants insofar as flow is concerned, with an extremely limited amount of forebay storage capacity.

It has not been demonstrated that the four-stage pump as proposed for a single-lift system would be as reliable as either the two-stage, double-suction pump or the single-stage pump, each on the appropriate alternative system. If a single-lift system is to be selected and built, many problems associated with its design remain to be resolved insofar as reliability is concerned.

Another factor which should be studied carefully is the comparative difficulty of providing adequate resistance to seismic accelerations in the case of (1) single-stage pumps, vertical or horizontal, (2) two-stage, double-suction pumps set horizontally, and (3) four-stage pumps set vertically.

The available evidence, both from model testing and observed prototype performance, indicates that for optimum reliability the relative efficiency would be highest for single-stage pumps designed for a three-lift or two-lift system, next highest for two-stage, double-suction pumps designed for a two-lift system, and lowest for four-stage pumps designed for a single-lift system; further, that the dependable efficiencies would be on the order of 93, 91, and 89 per cent, respectively. The model efficiencies are approximately 93, 90.5, and 88 per cent, respectively, based on tests to date of the three models being tested in the DMJM research and development program.

Comparative cost of power for pumping is a function of the over-all hydraulic efficiency of the respective alternative systems. Principal hydraulic losses other than within the pump units include friction losses in the penstocks, losses in the pump suction and discharge fittings, and losses arising from changes in velocity. If penstock diameters are optimized for minimizing combined costs of penstocks and pumping, the penstock losses will tend to be highest for single-lift systems. By appropriate discharge connections to reservoirs the fluctuations in water level can be accommodated without adding to the head loss. Complete hydraulic analysis of a considerable number of alternative systems shows that the two-lift and three-lift systems in general compare favorably with the single-lift systems in total dynamic head, and indicates no instance where the total dynamic head for a two-lift or three-lift system is more than 7 feet, 0.35 per cent, greater than for the single-lift system most favorable in this regard. Consequently, the pump efficiencies will control the comparative cost of power for pumping.

One of the aspects requiring attention is that success in achieving selection of the most promising concept would not of itself insure optimization of the system.

Competition among manufacturers of equipment under conditions where the lowest bid must be accepted has led on occasion to sacrifice of adequacy of design as well as quality of materials. This factor imposes a difficult burden in preparing specifications for, and in the inspection of, equipment which must be procured under these conditions, as past experience shows.

Considering the evidence at hand, and on the basis that a decision as to the pump lift system must be made in May, Metropolitan has taken the position of advocating, first, a three-lift system on the Pastoria Creek location and, second, a two-lift system on the Pastoria Creek or the ridge location, subject in each case to acceptable adaptation to environmental factors of topography, geology, stream flow, and streambed debris transport, among others. As definitive design of the selected system must be initiated promptly, Metropolitan has suggested that design of a three-lift system and of a two-lift system be commenced simultaneously, final selection to be made after the pump research and development program is further advanced.

For a two-lift system, the use of two-stage, double-suction pumps set in the horizontal position is supported by a formidable array of evidence. However, it is understood that the remainder of 1965 could be utilized for further exploration of the adaptability of single-stage pumps to this application, and this is recommended as an extension of the research and development program.

The following comments are directed to the question of pump capacity and number of pumps in each station of the system, including the question of standby capacity.

It is understood that the conveyance capacity of the approach canal to the Tehachapi pumping system and of the Tehachapi Tunnels will be 4,100 cfs, including an allowance of 7.5 per cent for outages. In this case, if the total installed capacity for the Tehachapi lift were 4,100 cfs, then whenever a pump is out of service, the pumping system would be unable to match the conveyance capacity of the adjacent aqueduct reaches. The question then arises, if the aqueduct reaches upstream or downstream might be shut down for repair at any time on their own account, what the consequences of such outages would be. While 7.5 per cent may be a greater allowance than justifiable for the aqueduct conveyance facilities, for purposes of illustration it will be assumed here that the adopted outage allowance for the affected reaches of aqueduct upstream and downstream from the Tehachapi pumping system is 7.5 per cent.

Proceeding from the foregoing assumption, it is evident that, after final-stage development and to the maximum extent feasible, the Tehachapi pumping system must be capable of delivering at least 4,100 cfs at any time. We have assumed that the affected conveyance facilities will be operative only 92.5 per cent of the time, and during the on-stream periods the pumping system must be up to full capability to lift 4,100 cfs in order to accomplish the total required annual delivery. It cannot justifiably be assumed that outage of pumping units for maintenance or repair will occur only during the outage of the affected conveyance facilities. The equipment outages may be both scheduled and unscheduled, as may the outages of aqueduct conveyance facilities; outage of one type of facility cannot be depended on to occur only at times when there is an outage on the other. In any event, orderly maintenance of pumping units requires the opportunity to have one unit at a time in each plant disassembled on a schedule of rotation. The only valid conclusion to be reached is that each plant in the Tehachapi pumping system must have a delivery capacity of at least 4,100 cfs plus a complete standby unit, if 4,100 cfs is the selected conveyance capacity of the affected reaches of the aqueduct. Regardless of the capacity of individual pumping units in relation to the required total delivery capability of a pumping plant, there should be one standby unit in addition to the number of units required for matching the conveyance capacity of the adjacent reaches of the aqueduct.

The only escape from the foregoing reasoning would be in the event that the outage allowance for the aqueduct conveyance facilities is greater than justifiable, or that outages of pumping units could be confined to periods of time when there is an outage of the affected conveyance facilities. If 7.5 per cent is too high an allowance, it would be appropriate to reduce it, but it would be inadvisable to make the assumption that outages could be confined to simultaneous occurrences.

On the foregoing basis, if the conveyance capacity of the affected aqueduct reaches is 4,100 cfs, then the pumping units for alternative systems might be as follows:

Single lift:	16 units @ 274 cfs, total installed 4384 cfs.
Two lift:	12 units @ 373 cfs, total installed 4476 cfs.
Two lift:	9 units @ 513 cfs, total installed 4617 cfs.
Three lift:	9 units @ 513 cfs, total installed 4617 cfs.

The alternative systems should be compared on the basis described above both as to costs and reliability, or on a basis equivalent in principle in the event some allowance other than 7.5 per cent is to be made for outages on the affected aqueduct conveyance facilities.

# # #

TEHACHAPI PUMPING PLANT

COMPARATIVE ANALYSIS OF LIFT CONCEPTS

PUMPS AND INTERFACE ELEMENTS

VOLUME I

.

COMPARATIVE ANALYSIS

April 1965

DANIEL, MANN, JOHNSON, & MENDENHALL
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TEHACHAPI RESEARCH & DEVELOPMENT PROGRAM

DANIEL, MANN, JOHNSON, & MENDENHALL

TECHNICAL ADVISORY BOARD

MEETING NO. 3

March 29, 30, & 31, 1965

MEMBERS PRESENT:

Ivan F. Mendenhall	-	Chairman
John T. Clabby	-	Member
Leslie J. Hooper	-	Member
Austin H. Church	-	Member
Peter Jaray	-	Member
David R. Miller	-	Secretary

Member Absent: - S. Logan Kerr

The Daniel, Mann, Johnson, & Mendenhall Technical Advisory Board was in session at DMJM Offices in Los Angeles on March 29, 30, and 31. A briefing of the progress to date, as presented in the draft of the four-volume Interim Report, "Tehachapi Pumping Plant, Comparative Analysis of Lift Concepts, Pumps and Interface Elements", was given to the Board by the Staff.

The Board has issued a statement, giving the following conclusions and recommendations:

1. Lift Concepts:

The Board has carefully considered the Staff's recommendations and comes to the following conclusions:

a. The comprehensive investigations at all major pumping facilities in Europe and the USA substantiate the suitability of each of the three pump types to its designated lift concept.

b. The three pump types now in the DMJM research and development program for the single-lift, two-lift and three-lift concepts are all feasible, according to the various studies which are presented in the interim report. They are all considered to be reliable and acceptable types of machinery for the Tehachapi Pumping Plant.

c. The Board accepts the relative ratings and comparisons made by DMJM; i.e., efficiency, reliability and costs.

d. The Board is aware of the fact that other features beyond the pumps, valves and motors may be important factors in the selection of the lift concept. The Board has, however, not considered any of such features as they are not part of the DMJM scope of work.

2. Model Test Results:

The Board has taken cognizance of the present state of the test program and of the efficiencies measured so far. Although these tests are preliminary and will be refined by forthcoming testing, the fact that the two-stage, double flow model shows better efficiencies than expected seems to be evident. The Staff has also expressed their opinion that the four-stage model does not necessarily present the optimum which could be achieved with this type.

The Board recommends, therefore, that an expanded program on the multi-stage pump be initiated to prove optimum efficiency data (unless the comparison of lift concepts, based on present data, leads to a conclusive decision). This expanded program should be the testing of two-stage, single-flow models at both Sulzer and Voith to clarify the influence of design and manufacturing differences which are apparent on the present models. This would take approximately two months.

All three model testing programs should be completed as previously approved. The information obtained from the three programs will result in the procurement of valuable information on the various pump types which will be of considerable benefit on future pump applications for the Department.

Additional efficiency information forthcoming from further model tests over the next 30 to 60 days should be made immediately available to all parties concerned.

3. Single-stage Pumps for Two-lift System:

The Board concurs with conclusions of the Staff as presented in the report, regarding the single-stage pumps which have been suggested for the two-lift system.

The head per stage is considered extreme as has already been substantiated by results of the wear test program.

The Board reaffirms its previous recommendation that in the present state of the art, the two-stage, double flow pump represents the best machine for the two-lift concept.

4. Wear Test Program:

Results obtained to date from the wear test program being conducted under supervision of the Department and DMJM at the Tracy Facility show that this program will be extremely beneficial for determining wear rates of various pump wearing element materials. They show that the water at this location is abrasive. Tests indicate so far that the wear is a function of both sample hardness and water velocity (for a given water quality) and the tests prove that the proper selection of materials will be extremely important.

It is recommended, therefore, that this program be given greater emphasis in the future, according to the recommendations as outlined in the report.

5. Motor Study:

The Board recognizes the fact that there is no applicable precedence for self-starting motors for any of the lift concepts. Furthermore, the Board is cognizant of the serious disagreement that has now become apparent between the various major motor manufacturers regarding the starting problems.

The Board feels that an accelerated research program on the motor and electrical system should be initiated and a detailed program should be recommended by the Staff as soon as possible. The detailed program should consider, among other activities, inclusion of:

a. Addition of at least one more TAB member well qualified in the heavy electrical motor field.

b. Paid studies to be handled by at least one motor manufacturer.

6. <u>Design Philosophy:</u>

Conservatism must be paramount in the final pump and motor design, with specifications requiring emphasis on minimum maintenance and maximum reliability - with first cost being secondary to the foregoing two items.

The R & D activities in the next program phase should be directed toward developing greater reliability and sustained efficiency in the final design.

CHAPTER 1

INTRODUCTION

The Department of Water Resources Engineers in February 1959 presented to the State Water Commissioners a report enumerating alternate routes for the aqueduct system to serve Southern California. The Commission, on February 26, 1959, adopted the Inland Route in concept as the most economical and feasible system. This Inland Route envisioned the utilization of a pumping plant to lift water over the Tehachapi Mountains, which would be the largest installation of its kind ever attempted. Department of Water Resources research indicated that very little precedence for a pumping scheme of this magnitude was available and it was decided that a research and development program was necessary.

Daniel, Mann, Johnson, & Mendenhall, in association with Motor-Columbus of Baden, Switzerland, was selected for this study, and on July 15, 1963, entered into a contract to program and perform the work which consisted of research and pump model testing to determine and analyze the feasibility, reliability, and efficiency factors for the system.

The work involved the investigation and analysis of all similar pumping facilities in the United States and in Europe, a review of the preliminary studies prepared by Department of Water Resources Engineers related to the pump lifts, and the initiation, administration, and performance of an investigation including a model pump test program to determine the most feasible and economical type of pumping facility. Consideration was given to such factors as efficiency, operation and maintenance characteristics, pumping plant equipment, and reliability and compatibility with related plant components such as motor, valves and penstocks. The program also involves conducting normal operation and transient behavior studies of the pumps to determine design characteristics for the pumps and appertenant equipment including analysis of three quadrant pump characteristics and water hammer analysis of the penstock system, discharge valve operating time, pump starting and stopping and emergency operation conditions, pump motor thrust bearing, foundation anchorage, space requirements and pump failure problems. A final report is to be submitted to the State which shall include a draft of the technical portion of the specifications for procurement of the prototype pumps.

This interim report will give a resume of the work done by Daniel, Mann, Johnson, & Mendenhall on the Tehachapi Pumping Plant to date.

Early in 1964, following the preliminary report by DMJM, the pump analysis was limited to three models, i.e., (1) four-stage, single-flow for single-lift scheme; (2) two-stage, double-flow for two-lift scheme; and (3) single-stage, single-flow for three-lift scheme.

The model pump testing program is proceeding on the three pump types selected, and preliminary testing has been completed.

A Wear Test Facility has been constructed in order to determine the erosion and corrosion effects on pump parts using water which the Tehachapi Pumping Plant is expected to handle in actual operation. It is located at the Tracy Fish Screen Intake Facility at the head of the Delta-Mendota Canal. The construction of the facility and installation of equipment was started in August, 1964, and actual testing has been underway for four months. Preliminary results are included in this report and they indicate that the wear testing program will contribute valuable data.

A well documented report was issued by the Department of Water Resources in September 1964, entitled "Technical and Economic Feasibility of Single-lift, Two-lift, and Three-lift Systems; Tehachapi Pumping Plant."

This interim report by DMJM supplements the above report, as it pertains to the pumps proper and the interface items. It includes the input from those phases of the investigations made in 1964 that will influence the selection of the final lift concept.

Wherever possible, the recommendations have been documented with data from existing installations, from major manufacturers, from experts in the field, and from pertinent literature. Model test data with currently available results and conclusions are included.

A study of the overall system operation, construction and design and cost analysis is not included in the contract, and this work is being performed by the Department of Water Resources. Only such items which influence the pumping plant proper are included in the report; however, a cursory review of the civil and topographic features covering the complete pumping facility has been made, and comments are included herewith.

A summary giving the organizational and biographical background of DMJM, their associates and consultants, is given in Volume IV.

CHAPTER 2

FINDINGS, CONCLUSIONS & RECOMMENDATIONS

DMJM has made an exhaustive investigation of all pump types which should be considered for the three different lift concepts. This included visits to both European and American pumping plants considered pertinent, as outlined in Volume II, Chapter 2 and Volume III.

This study also included comments from major pump manufacturers and consultants, and the pump types finally selected for each of the three lift concepts are all considered suitable for the ultimate project.

A. LIFT CONCEPTS:

Regardless of the pump type finally selected, be it a single-stage, two-stage or four-stage pump, there is no doubt whatever that the pump industry will be able to design and build pumps for Tehachapi that will be reliable and will give satisfactory service over the next 50 years.

The efficiencies predicted by the model test program and the reliability factors documented in the reliability sections of this report, together with the cost estimate of the pumps, motors and valves, will provide an accurate basis for evaluation and final analysis of the three different lift concepts.

A decision on the final lift system most acceptable after consideration of all factors concerning the complete project, may then be made.

B. PUMPS:

It seems pertinent to discuss here the various pump types considered for the three different lift concepts for evaluation of their respective merits.

A basic factor in evaluating pump types was the requirement that long time operational experience data of comparably sized units be analyzed rather than assembling data by extreme extrapolation from short term experience and/or smaller plants. It is for this reason that the exhaustive investigation was made of existing large European and American pumping plants. This criteria includes horsepower rating of units, capacity of units, head of units including maximum head per stage, and operating speed of units.

All types selected are represented by units of comparable size which have been in satisfactory operation for a considerable length of time.

1. Pumps for Single-lift Concept

The pump tentatively selected by DMJM for this lift concept is a four-stage pump operating at 600 rpm, and requiring a motor rating of approximately 75,000 HP. As has been documented in Volume II, Chapter 2, Section D, there is precedent for pumps of this type operating satisfactorily in Europe, and typical installations exist at Lunersee, Etzel, Ponale, Tierfehd, and Motec.

There are five units in satisfactory operation at the Lunersee plant, having a horsepower rating of 58,000, and operating at a higher speed of 750 rpm against a higher head of 3150 ft. It has been pointed out that these pumps have a lower specific speed than the units selected for Tehachapi; however, as discussed in Volume II, Chapter 3, Section B, this factor is not pertinent and should be entirely ignored when considering operating experience and reliability. The Lunersee pumps operating at comparable horsepower and higher speed are, therefore, considered a good comparison for evaluating this type of unit for Tehachapi.

This type of pump is of a more complex design than those selected for the other two lift concepts and its efficiency will be somewhat lower. It has a balancing arrangement which must absorb the full pressure drop; however, experience at Lunersee indicates that the multiple labyrinth incorporated in this design has a favorable reliability record, probably due to the fact that the velocity through the individual labyrinth will be low.

On the positive side, this pump will have a relatively low head per stage of 488 ft. which is well within the present experience of satisfactory long life pump operation. Logistical advantage accrues to the single lift concept in that this concept permits optimum pump plant accessibility. This concept produces the greatest operational flexibility which, evaluated quantitatively, indicates a 6.25% loss of system capacity per pump compared to an 11% loss per pump in the other concepts.

The submergence that will be required for this type of pump is within the proposed limits of excavation for the pumping plant construction.

2. Pumps for Two-lift Concept

The pump selected by DMJM for this lift concept is a two-stage double-flow pump operating at 600 rpm and requiring a motor rating of approximately 68,000 HP. As documented in Volume II, Chapter 2, Section D, there is ample precedent for this type of unit in Europe, larger units operating at higher heads having been in satisfactory operation for many years.

This unit has a moderate pressure per stage of 488 ft., is hydraulically balanced, requiring no balancing disk and thus has no wearing parts in the pump exposed to a differential pressure higher than 488 ft. The design is approximately as complex as that for the four-stage pump; however, the estimated efficiency is higher.

The submergence required for this unit is approximately the same as for the four-stage pump discussed under 1., and similarly should present no problem in the installation.

The fact that two separate pumping stations would be required for this type of unit must be evaluated and this has been considered in the reliability established in the Reliability section.

There is no question that if the two-lift concept should be finally adopted, this type of pump would be an optimally efficient selection.

3. Pumps for Three-lift Concept

The pump selected by DMJM for this lift concept is a single-stage, single-flow pump operating at 514 rpm and having a motor rating of approximately 46,000 HP.

This unit has extensive precedent in American practice, although there is little precedent for pumps having an equivalent head of 650 ft. However, many pumps are in operation with satisfactory efficiency and reliability records with heads up to 500 ft. and extrapolation to a 650 ft. head should be acceptable.

This concept produces the highest stage head considered, and was one of the reasons for the establishment of the wear test program now being conducted at Tracy Fish Intake, where test samples are exposed to equivalent heads for long periods, handling water having qualities similar to that expected for the Tehachapi Crossing. Preliminary results of this wear test program are included in Volume II, Chapter 10.

This type of unit requires higher submergence than the other two types; however, due to the axial inlet, the performance which will be established by the model test program conducted may indicate that the required submergence will not be significantly greater than for the other types.

Finally, this concept produces a lower concept effectiveness than the other concepts considered.

4. Single-stage Pump for Two-lift System

It has been suggested that a single-stage pump be considered for the two-lift system as an alternate to the two-stage double-flow pump now in the program. There is no question that a single-stage pump having a power rating of approximately 68,000 HP and working against a head of approximately 975 ft. can be built from the hydraulic standpoint. Smaller pumps operating against a higher head have been built and are in satisfactory operation. The problem, however, is entirely different for pumps handling raw water for heads in the range of 1,000 ft. in one-stage. As is pointed out in Volume II, Chapter 2, Section D, there is no precedent for this type of unit in either European or American practice, and if the criterion as established is to be followed, long operating experience on similar units should be available for comparison. No experience is available to establish the possible wear rate for such units, and it is likely that they would require excessive maintenance. Model tests for this type of unit would lack long term operating experience.

This unit will also require considerably more submergence than any of the other types. If a relatively low specific speed of 1470 is selected which is equivalent to an operating speed of 514 rpm, the unit would still require a minimum of about 85 to 90 ft., even when considering the "no shaft through the inlet" condition, for satisfactory cavitation performance. The efficiency of this unit would probably be lower than the other types being considered because of its low value of specific speed. Its efficiency might still be higher than that of the four-stage pump, however.

This pump is not included in the DMJM study.

C. RECOMMENDATIONS FOR CONTINUATION OF DMJM RESEARCH PROGRAM:

It is anticipated that the final lift concept to be adopted will be established, based on the input from the DMJM study into the overall study being conducted by the Department of Water Resources, before the projected model testing program is completed.

DMJM recommends that the following Research and Development program be adopted for the balance of the contract term:

1. Continuation of Model Test Program

All three model testing programs should be completed in accordance with the original contracts. The information obtained from the three programs will result in the procurement of valuable information on the various pump types which will be of considerable benefit on future pump applications for the Department.

In order to obtain a design comparison for a multistage pump, it is recommended that an extension of the program be made to test two stage, single flow models at both Sulzer and Voith. The two stage models would be assembled for the most part from existing components. These tests would clarify the influence of design and manufacturing differences that are apparent on the current models being tested by these two firms.[1] This effort would require approximately two months and would cost an estimated additional $25, 000 to $30,000.

2. Model Comparison Tests

After the completion of the contract testing program at the manufacturing laboratories, it is recommended that the three models be sent to the National Engineering Laboratory at East Kilbride, Scotland for comparison testing under the supervision of DMJM. The new 5000 HP dynamometer which is being installed at this facility will permit testing of the models at approximate prototype head, although the two stage, double flow model will be limited to a test head of approximately 90% due to casing strength limitation.

These tests will prove the validity of the test results obtained at the manufacturers laboratories on an exact comparison basis.

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[1] See Volume II, Chapter 3 - Diffuser-Return design discussion.

3. Wear Test Program

The wear test program at the Tracy Fish Intake facility should be continued for three additional seasons to establish the wear pattern of various pump wearing ring materials when exposed to aqueduct water at different heads. These tests will give valuable information on the effect of the seasonal variations in the water quality on various construction materials.

It is recommended that the program effort and scope be increased to achieve the following objectives:

a. Additional personnel be provided for more frequent monitoring of the facility. Frequent equipment failure has occurred due to the fact that the facility is only visited once a day. More frequent facility visits are necessary and continuous daily attendance would be beneficial.

b. Two additional rotating testers be built to produce pump evaluation data idiosyncratic to this program. To accomplish this, the testers should be designed by DMJM and built by the Department.

c. Additional personnel should be assigned by the Department for water analysis and data processing, including specimen weighing, and so forth, in order to have the results available sooner.

d. A complete set of mechanic's hand tools should be provided at the facility along with an adequate supply of spare parts for the machinery.

4. Prototype Design Study

The prototype design should be finalized using the input from the completed model test program, and also the information obtained from the plant visits.

The material selection for the various parts of the prototype pumps should be established, using as input the results obtained from the plant visits and the wear test program.

Final specifications for the prototype pump should be prepared. The minimum acceptable efficiency level as established by the model tests should be specified.

5. Motor Study

As soon as the prototype pump type and size has been established a more exhaustive investigation of the driving motors, their starting characteristics and controls should be initiated. As outlined in Chapter 4 of Volume II, there are serious differences of opinion regarding the most desirable starting methods between major motor manufacturers. Therefore, additional studies, including possibly some experimental work, are suggested before this matter can be satisfactorily resolved. The possibility of alternate starting methods to across-the-line starting should be further explored.

6. Valve Study

The valve study should be concluded for the selected lift concept and final specifications prepared.

7. System Study

A system study, including a vibration study, should be made of the combined motor-pump assembly. This should include pump and motor supports, methods for assembling and disassembling and preliminary plant layouts. This should also include final cost estimates.

8. Study of Intake Piping

A study should be made of the piping between the forebay of the first pumping station and the pumps. Model tests are recommended for inclusion in specification for the prototype pumps to assure a smooth flow pattern in the pump approaches.

9. Hydraulic Transients Study

A comprehensive surge and water hammer study should be made for the final lift concept as soon as the discharge pipes and their alignment has been determined.

10. Reliability Study

The reliability study should concentrate on the absolute reliability and availability of the concepts and on the design optimization which reliability techniques provide. Comparative results as given in the interim report should be refined by analysis of repair time probability distributions, refinement of scheduled outages, revision of data processing with regard to wear test results, and application of continued literature search disclosures. Cost-effectiveness calculations for the wear life, efficiency loss life and optimum repair and maintenance schedule life of various pump element designs, materials and system configurations should be evaluated to provide optimum design and maintenance planning criteria. Effects of varying plant overcapacity and planned repair techniques should be investigated.

11. Management Control Program - PERT

The PERT Management Control Program should be continued to the conclusion of the program. This program is currently the management tool which most rapidly and thoroughly provides program schedule progress information. This information most readily enables management to plan construction completion schedule corrective action as required. Additionally, the prompt distribution of periodic PERT reports to all contributing organizations enables each to continually re-evaluate its individual contribution to the total program.

CHAPTER 3

PROJECT APPROACH

In order to ensure the utmost in integrity, reliability, and confidence in the final solution of a program, DMJM has developed an approach based on sound engineering principles, formulating and evaluating the various scientific and analytical techniques and the budgeting and assignment of the proper engineering disciplines. The approach adopted by DMJM for the solution of the Tehachapi pump program included the development of a Program Analysis Diagram (PAD) in the planning stages of project performance. The Program Analysis Diagram was stochastic in nature and provided the relative order in which tasks, subtasks, and elements of the program. The Program Analysis Diagram provided the basic mechanism for developing a PERT management control network.

Within the PERT framework for the development of the design parameters and specifications for the pumps utilized at the Tehachapi Crossing, DMJM has identified and is performing the following technical efforts:

A. <u>Pump Research Study</u>

This is a study to monitor, validate, review, and interpret model pump test results. This study is limited to the pumps and does not include other elements of the pumping station. DMJM, Motor-Columbus, and consultant personnel are performing this study. Study elements are:

1. Prototype design analysis.

2. Prototype efficiency analysis.

3. Specific speed as it affects efficiency, cavitation, and performance.

4. Study and survey of causes and means of avoiding undue wear, cavitation damage, and loss of efficiency.

5. Study of safety factors and materials in pumps.

6. Study of vibration and stress problems.

7. Cost and weight determination.

8. Investigation of existing pump installations with regard to failure and behavior experience, and maintenance and operation problems.

9. Study of resonant effects from pressure oscillations in discharge piping connections.

10. Study of pump mounting and support requirements.

B. <u>Hydraulic Transients Study</u>

This study is being conducted to define the surge problems imposed on the pumps in order to more completely predict operating conditions. Study elements are:

1. Starting conditions.

2. Emergency operation.

3. General surge analysis.

C. <u>Reliability Study</u>

The reliability study is being performed to provide data for selection of the optimum pump concept, to provide design parameters, and to establish a basis for pump operation and maintenance. Although the pumps receive primary attention in this analysis, a summary of the effects of other pumping station elements, such as, valves, motors, penstocks, switch gear, transformers, gates, and plant auxiliaries have been included. Study elements are:

1. Field experience survey.

2. Data tabulation.

3. Normalization of data.

4. Component life predictions.

5. Reparability predictions.

6. Concept availability predictions.

7. Concept reliability predictions.

8. Design effects analysis.

9. Maintenance schedules planning.

D. Wear Test Program

1. Tester design and construction.

2. Facility design, construction, and installation.

3. Seasonal stationary tests.

4. Seasonal rotary tests.

5. Analyses of test results.

6. Materials and design recommendations.

E. Motor Study

The motor study was conducted to establish feasibility, pump maximum speed and data for the reliability study. Study elements are:

1. Analysis of motor requirements for lift concepts.

2. Study of auxiliary electrical equipment requirements.

3. Study of starting problems.

4. Conferences with motor manufacturers on motors, thrust bearings, and couplings, as related to complete pumping unit.

5. Survey of size, weight, and space requirements.

6. Assemblage of cost estimates.

7. Collection of failure and behavior data as they affect pump operation.

F. Valve Study

This study was conducted to provide data on size, cost, and operating characteristics of valves suitable for the three plant concepts. In addition, data were gathered for the reliability study. Valve study elements are:

1. Develop criteria and valve requirements.

2. Study operating characteristics and problems of starting and transient conditions, including valve operating controls.

3. Discussions with manufacturers regarding size, weight, and space requirements.

4. Assemblage of cost estimates.

5. Failure and behavior study of existing valve installations.

G. Allis-Chalmers/Sulzer Model Pump Program

This contractor was selected to perform the research and design program on four-stage, single-flow pumps for the single-lift concept. DMJM analyzed their program and established the following elements as major events in the performance of this portion of the program:

1. Fundamental design of prototype pump, scale down, and design of model.

2. Fabrication of model and preparation of the laboratory setup.

3. Installation of model in the laboratory and calibration of instruments.

4. Tests on the model.

5. Preparation of final report.

H. Baldwin-Lima-Hamilton/J. M. Voith Model Pump Program

This portion of the program will provide data on the two-stage, double-flow pump for use in the two-lift concept. Study elements are the same as those of section G, above.

I. Byron Jackson Pump Program

This program is intended to provide optimally valid design criteria for a single stage, single suction pump as used in the three lift concept. Study elements are the same as those of section G, above.

CHAPTER 4

STUDIES OF LIFT CONCEPTS

A. HISTORY:

As early as 1955, serious studies were started by the State and by consulting firms on the feasibility of the Tehachapi Crossing Pumping schemes. Subsequent investigations and reports covering the period to the end of 1963 are enumerated in Chapter IV of DWR's preliminary report.[1] As is well documented in that report, there has been a tremendous amount of investigative work performed by various State authorities, consultants and manufacturers for the purpose of selecting the most suitable configuration of the overall arrangement.

The earlier reports favored, in general, a single-lift arrangement for the Tehachapi Crossing.

However, since the Tehachapi Crossing Consulting Board meeting of October 1963, at which a firm selection of a ridge route was made, alternate schemes were proposed by others, utilizing the ridge and Pastoria Creek alignments and using multi-lift systems.

It was mutually agreed after the October 1963 meeting of the TCCB, therefore, that the single-lift, the two-lift, and the three-lift arrangements for the pumping plants should be further evaluated and the model testing program was to include models suitable for all three lift concepts.

Investigations conducted by the Department of Water Resources Engineers and by engineers retained by the Metropolitan Water District resulted in lift systems and pump recommendations differing in their appraisal of reliability and efficiency.

DMJM, after the Technical Advisory Board meeting in February 1964, proposed a model testing program that was pointed toward investigating all types of pumps considered suitable by both parties. Subsequently, the program was reduced from the (5) models proposed to (3) models, one for each lift concept.

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[1] "Preliminary Report of Technical and Economic Feasibility of Single-Lift, Two-Lift, and Three-Lift Systems; Tehachapi Pumping Plant", Department of Water Resources. State of California, Sacramento, 1964.

It is repeated that the DMJM contract and this report pertain to the pumping plants and interface items including motors, valves, pumps, surge problems, and so forth, only, and do not include a study of the overall system.

B. OVERALL PLANS AND SKETCHES OF THE DIFFERENT LIFT CONCEPTS:

As explained in Chapter 5, "Design Parameters & Restraints", it has been established for the purposes of this report that the "ridge alignment" is to be used for all three different lift systems considered.

Fig. 4-1 shows the general location of the Tehachapi Crossing in the overall aqueduct scheme.

Fig. 4-2 shows the location and the vicinity maps of the Tehachapi Crossing and also the proposed ridge alignment.

Fig. 4-3 (DMJM Dwg.) shows two alternate arrangements for the single-lift system, both with underground discharge lines.

Fig. 4-4 shows two alternate arrangements for the two-lift system, one with surface discharge lines and surface plants, and one with underground discharge lines and an underground plant for the upper pumping station.

Fig. 4-5 shows two alternate arrangements for the three-lift system, one with surface discharge lines and surface plants, and one with underground discharge lines and underground plants for the upper two pumping stations.

C. DISCUSSION OF DIFFERENT LIFT CONCEPTS

The following numbered sections review project element positioning from the standpoint of its influence on pump operation emphasizing particularly pump reliability. Some general factors with respect to project element positioning are noted. Associate Consultants, Motor-Columbus, Baden, Switzerland, concur.

1. Surface Versus Underground Discharge Lines

California lies on the earth's largest and most active seismic zone, the Pacific Ocean Periphery Seismic Zone. This project spans one of the major zones of differential seismic movement in the State and is

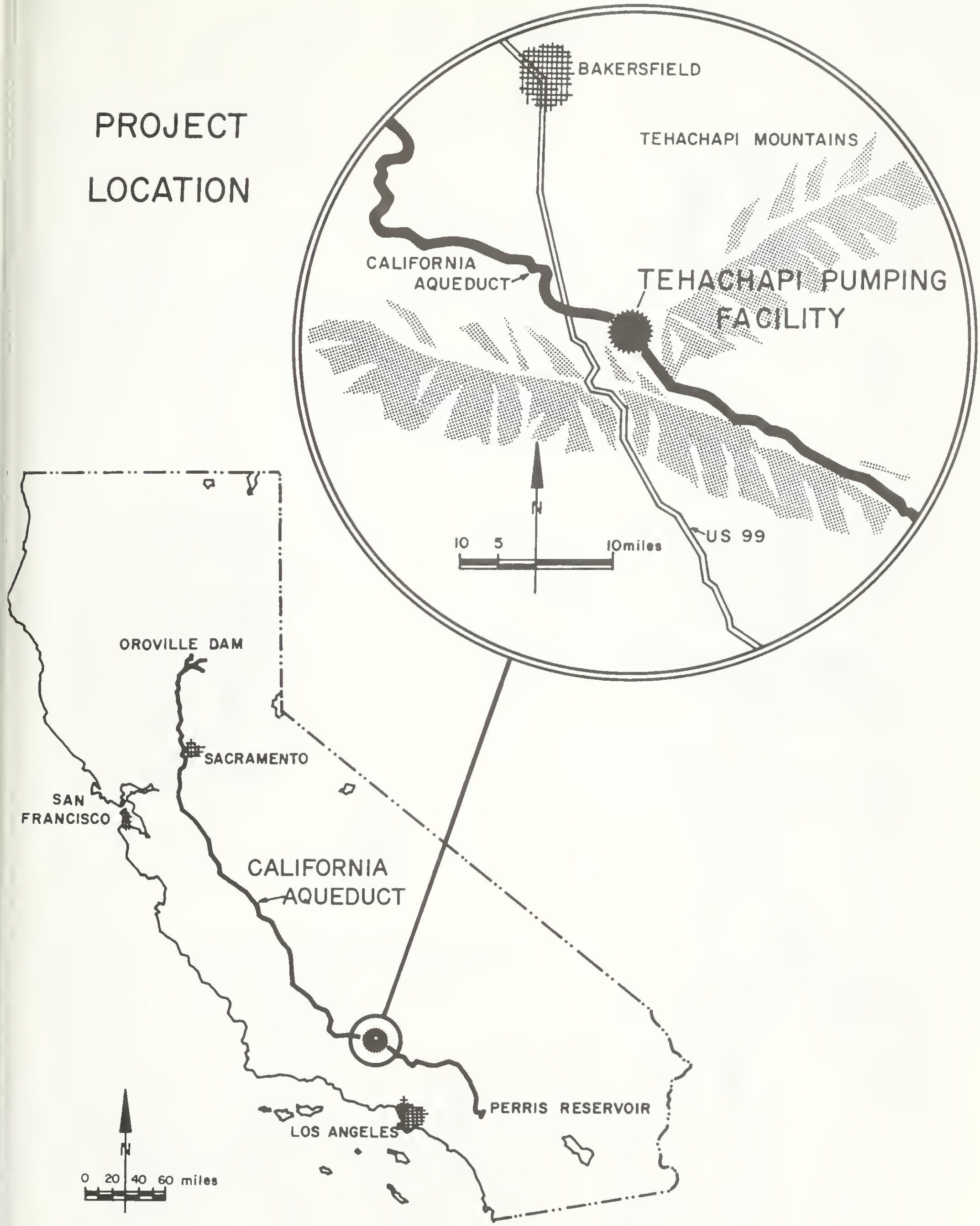
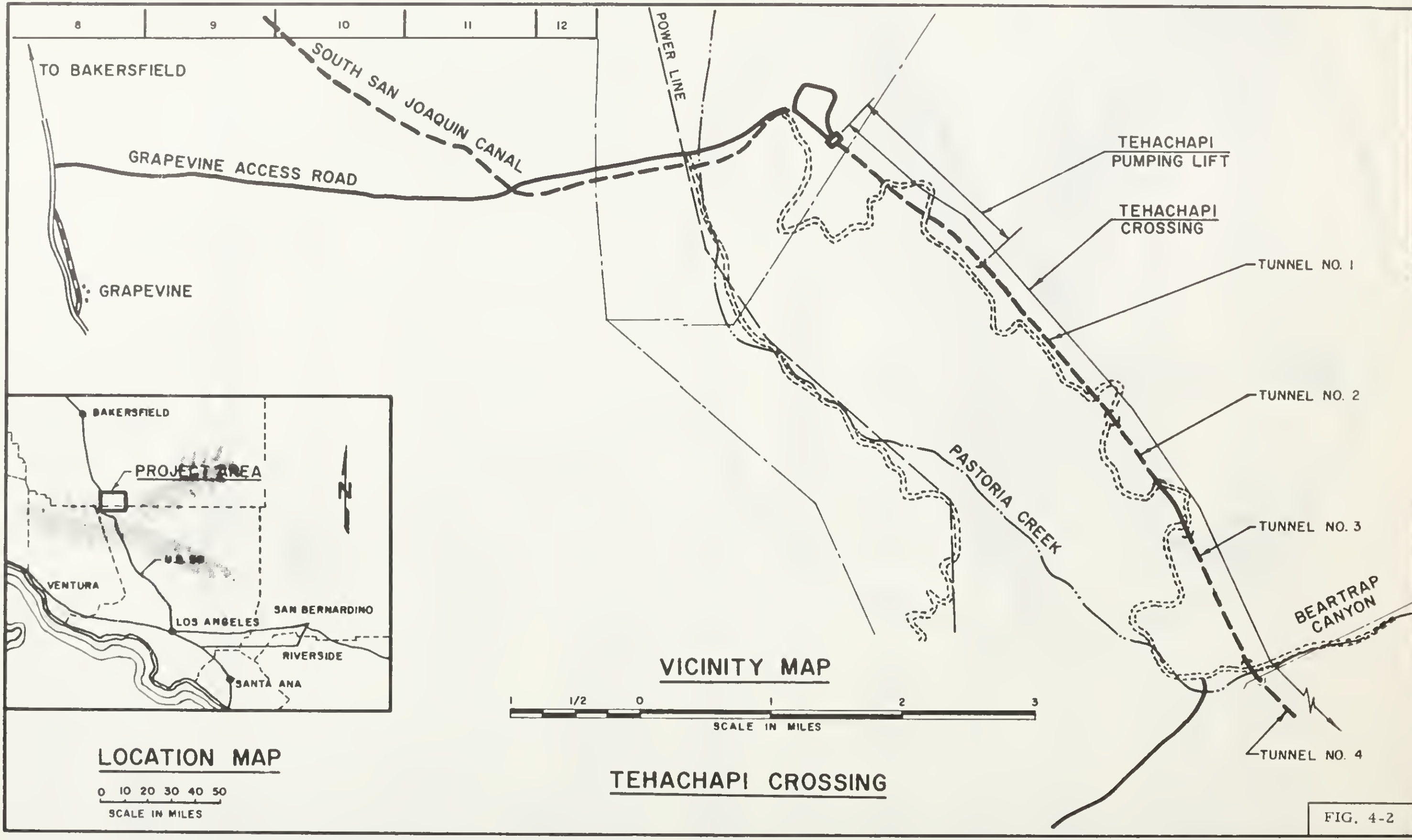

PROJECT
LOCATION
BAKERSFIELD
TEHACHAPI MOUNTAINS
CALIFORNIA
AQUEDUCT
TEHACHAPI PUMPING
FACILITY
N
10 5 10miles
US 99
OROVILLE DAM
SACRAMENTO
SAN
FRANCISCO
CALIFORNIA
AQUEDUCT
PERRIS RESERVOIR
LOS ANGELES
N
0 20 40 60 miles

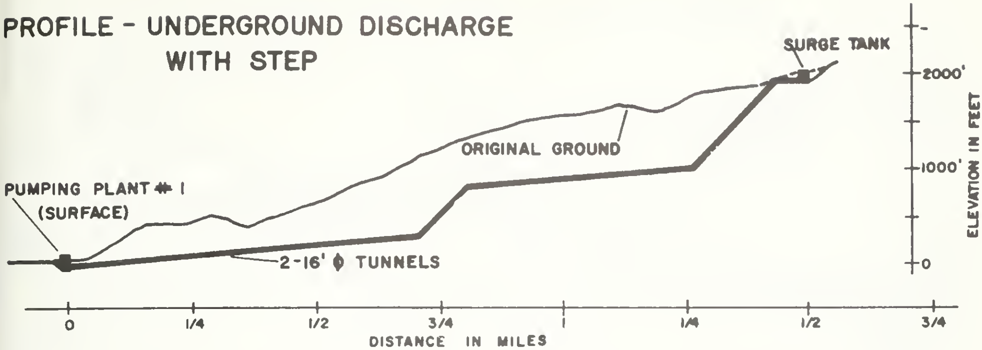
FIG. 4-1

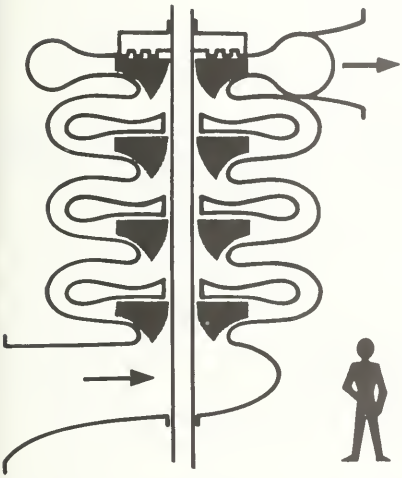


TEHACHAPI CROSSING

FIG. 4-2

SINGLE LIFT SYSTEM





PUMP TYPE	FOUR STAGE, SINGLE SUCTION
SPEED	600 RPM
FLOW	140,000 GPM
HEAD	1951 FEET
NO. OF STATIONS	1
NO. OF PUMPS PER STATION	16

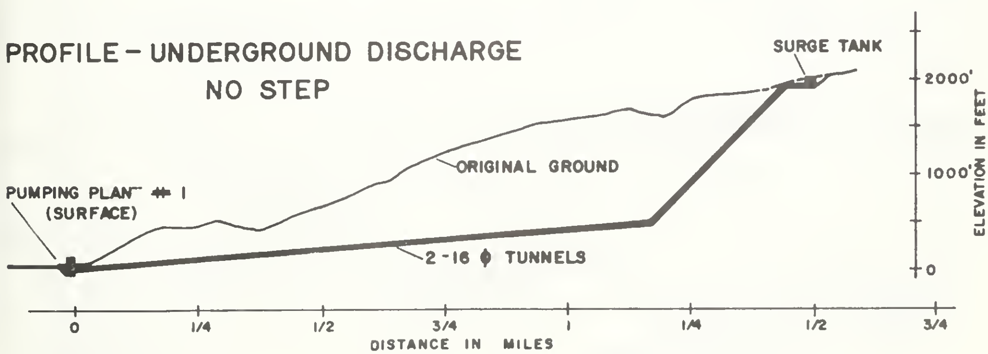
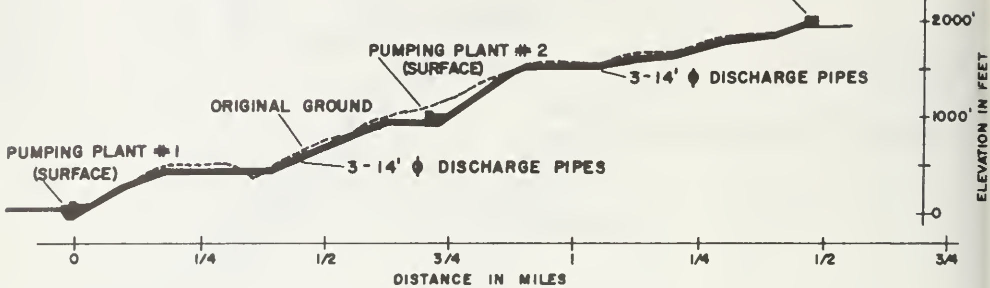
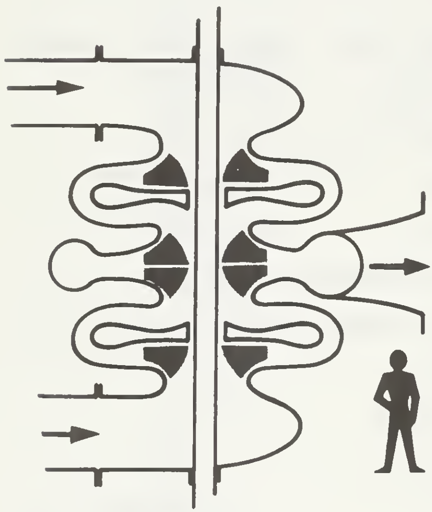


FIG. 4-3

TWO EQUAL LIFT SYSTEM

PROFILE - SURFACE DISCHARGE LINES





PUMP TYPE	TWO STAGE, DOUBLE SUCTION
SPEED	600 RPM
FLOW	249,000 GPM
HEAD	976 FEET
NO. OF STATIONS	2
NO. OF PUMPS PER STATIONS	9

PROFILE - UNDERGROUND DISCHARGE LINES

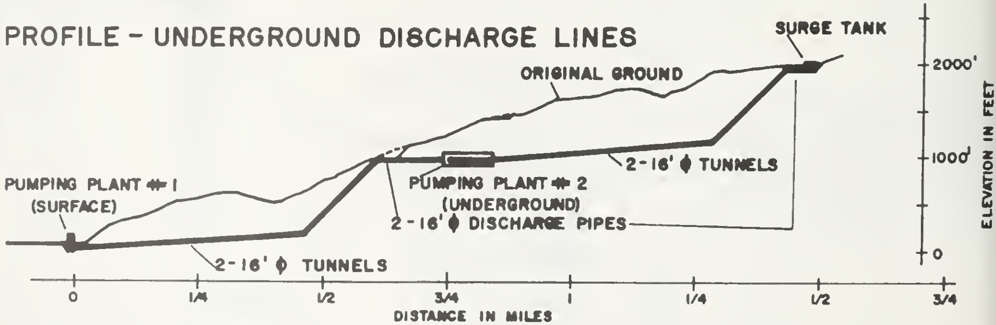
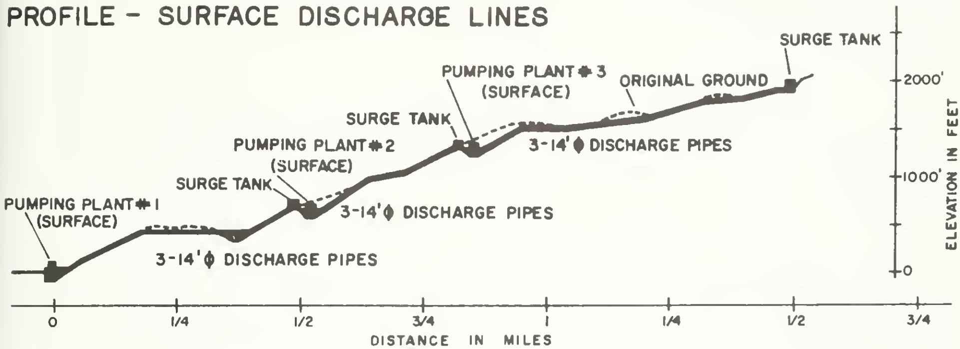
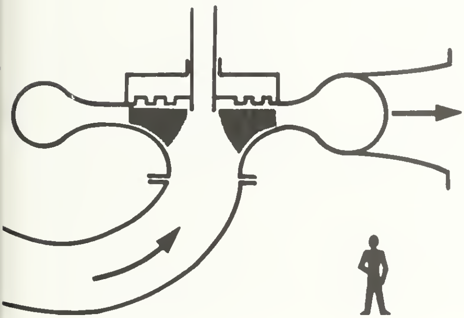


FIG. 4-4

THREE EQUAL LIFT SYSTEM





PUMP TYPE	SINGLE STAGE, SINGLE SUCTION
SPEED	514 RPM
FLOW	249,000 GPM
HEAD	650 FEET
NO. OF STATIONS	3
NO. OF PUMPS PER STATION	9

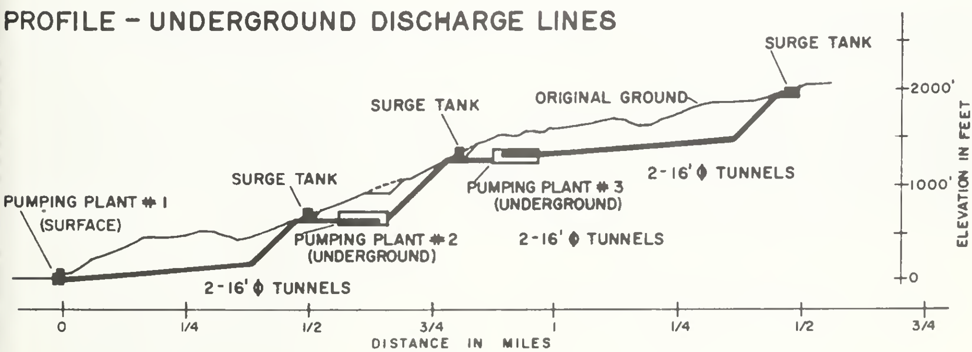


FIG. 4-5

consequently particularly subject to the hazards of seismic disturbance. Additionally, the project site is in a mountain range and construction will include all of the difficulties and hazards usually occuring with mountain construction.

Typically, the geometry of location of this type project is determined by decision based on engineering analysis including economic, financial and intangible analyses. These analyses include complete consideration of anticipated construction problems, reliability and geologics, including seismics.

From the standpoint of total discharge line failure, such as could result from local seismic disturbance of large magnitude, the resulting damage to pumps will usually be much greater with surface conduits than with underground conduits. This occurs because of the frequent location of the surface conduits (discharge lines) and upstream surge tank or reservoir water mass above and in line with the pump while conversely, underground conduits seldom occur in this position. For this reason, from the standpoint of project pump comparative analysis, underground discharge lines are preferable over surface discharge lines for all lift concepts considered.

2. Surface Versus Underground Stations

Usually the type and location of pump stations in projects of this character are determined by engineering analysis as mentioned in section 1, above. The overall cost may be influenced by the fact that with an underground station a shorter discharge line may be possible than with a surface station. Short discharge lines are also preferable with respect to water hammer and it is, consequently, frequently advantageous to locate the underground station deep in the mountain. This position is also frequently advantageous for geotechnical reasons.

Underground pump stations offer more pump submergence flexability than do surface stations. Deeper setting of the pumps is possible at less additional cost than with a surface plant. Furthermore, high and steep rock cuts with the resultant slope stabilization problems that frequently occur are usually required with surface plants while seldom required with underground plants. This type problem is particularly prevalent with intermediate plants in multiple lift schemes. For these reasons, from the standpoint of project pump comparative analysis, underground stations are preferable over surface stations for all lift concepts considered.

It is believed that based on the factors mentioned above, unless clearly adverse geologic conditions prevail, underground construction of this pumping station and interface discharge lines will score more favorably when subjected to complete engineering analysis including full geologic evaluation than will a surface plant with surface discharge lines. Consequently, further evaluation of the location of this station is recommended.

3. Discharge Lines and Discharge Line Alignment

As mentioned in Section 1, above, underground discharge lines are considered to be the safer solution in general. If geological conditions are unfavorable and surface lines must be chosen, provisions should be made to divert leakage or overflow water without causing further damage. The pipe alignment should be such that the pumphouse is not placed right in line with the discharge line, i. e., right in the course of the water in the event of a penstock rupture.

4. Flood Considerations

While gravity drainage is possible at the intermediate stations in the multilift concepts, major parts of the first pumping station are in any case situated below suction water level. Gravity drainage is not feasible at this station with any lift concept, thus requiring special safety drainage. Water turbine driven emergency drainage pumps should be considered.

CHAPTER 5

DESIGN PARAMETERS & RESTRAINTS

A. BASIC CRITERIA:

During meetings of the Tehachapi Crossing Consulting Board, meetings of the Technical Advisory Board, and subsequent meetings with Department Engineers, the following guidelines were established for the project at the beginning of 1964:

1. One, two, and three lift systems are to be considered for the Tehachapi pumping station, and all three concepts are to be thoroughly investigated.

2. The alignment of the discharge pipeline is to be on the Tehachapi Ridge (see Fig. 4-2).

3. A surface plant is the only one to be considered for the first pumping station, located next to the forebay.

4. A total plant capacity of 5000 cfs shall be considered the design flow for the system.

5. The normal static lift for the pumping conditions is from elevation 1239 ft. to elevation 3168 ft., or a static lift of 1929 ft., and the resultant total dynamic head is approximately 1951 ft., subject to minor corrections for the different lift concepts.

6. The motor size for each pump is to be limited to approximately 75,000 HP, and the running speed should not be more than 720 rpm.

7. Only three models are to be tested, one for each lift concept.

B. BASIC PUMP DESIGN PARAMETERS:

1. Pump Selection and Pump Operating Conditions

The Technical Advisory Board, at its February 1964 meeting, recommended considerations of the prototype pumps as shown in Table 5-I, which considered seven (7) pump types and required a total of five (5) models for the testing program. At this time, a total flow rate of 3510 CFS with a total dynamic head of approximately 1951 ft. was being considered.

Plant Concept	No. of Pumps	Flow CFS per Pump	No. of Stages	Total Head -Ft. per Pump	Head per Stage - Ft.	Speed RPM	Specific Speed	Submerg. Ft.	Model in Program	Model Phase No.	Type of Pump	Item No.
Single	12	292.5	4	1951	488	600	2095	53	**	1A	M.S.	1
Lift	12	292.5	2	1728	864	900	2048	113 (25)			S.S.(2) (B)	2
Two Plant	6	585	2	975.5	488	400	1970	46	**	1B	B-to-B	3
Multilift	6	585	2	975.5	488	600	2090	53	**	1D	D.F.	4
	6	585	2	975.5	488	400	1970	46			S.S.(2)	5
Three-Plant	6	585	1	650	650	400	1600	46	**	IC-2	S.S.	6
Multilift	6	585	1	650	650	514	2060	77	**	IC-1	S.S.	7

EXPLANATION OF SYMBOLS:

M.S. - Multistage
B-to-B - Back-to-Back
D.F. - Double Flow

S.S. - Single Stage - Single Flow
S.S.(2) - 2 Single Stages in Series
(B) - With Booster Pump

NOTE: MODEL TESTS FOR ITEM 7 WILL ALSO BE APPLICABLE TO ITEM 2 AND ITEM 5.

TABLE 5-I MODEL TEST PROGRAM
RECOMMENDED BY TECHNICAL ADVISORY BOARD - FEBRUARY 1964
TOTAL FLOW 3510 CFS

Subsequently, the flow rate was increased to 5000 CFS and the research program was limited to three pump types, one for each of three different lift systems. Data for the selected types is given in Table 5-II.

It should be pointed out that the selection of these three pump types was made after screening all available types, after consulting major pump manufacturers, studying existing installations all over the World, and after consulting a number of well known experts in the pump field.

The number and size of the units for the different lift concepts were selected by the Department after a comprehensive study. The selection was consistent with the requirements of flexibility for the lift systems, of eliminating part load operation even during the initial phases, and with the basic design parameters and restraints enumerated under paragraph 5A.

2. Pump Arrangement

If, as outlined in paragraph 5A, the first pumping station must be a surface plant, it follows that the pump arrangement should preferably be of the vertical shaft type. This is due to the fact that all pumps considered require a submergence in excess of 50 feet and it is undesirable to place the motors so far below tail water level where they may be subject to accidental flooding. This arrangement is also in line with the majority of large pump and water turbine installations in the United States, which have a history of good operation.

It must be pointed out, however, that the majority of large pump installations in Europe (see the report in Chapter 2 of Voluem II) are installed with a horizontal shaft and this arrangement is much preferred by the operators. Therefore, the horizontal shaft configuration should not be eliminated from consideration at this time, particularly for the upper stations of the two and three-lift systems, where accidental flooding is less likely to occur.

The final decision on the most desirable arrangement can wait until the lift concept is selected. It does not affect the model testing program, as all models are being tested in a horizontal shaft position.

3. Pump Efficiency

The importance of procuring pumps for the project having the maximum efficiency consistent with long life and good reliability is selfevident. The Department has estimated that a difference of 1% in pump efficiency for the 5000 CFS System has a present worth value of approximately $9, 000, 000.

TABLE 5 - II

TEHACHAPI MODEL TESTING -- PROTOTYPE AND MODEL DESIGN CRITERIA

DMJM SUBCONTRACT		637-1-1A	637-1-1B	637-1-1C
MODEL FIRM		Allis Chalmers	Baldwin-Lima-Hamilton	Byron Jackson
LIFT CONCEPT TYPE		Single Lift	Two-Lift	Three-Lift
PUMP TYPE		Four Stage Single Suction	Two Stage Double Flow	Single Stage Single Suction
PROTOTYPE	H (FT)	1951	975	650
	Q (CFS)	312.5	555.6	555.6
	N (RPM)	600	600	514
	N_s (Specific Speed)	2160	2040	1990
	No. of Units/Station (Based on 5000 CFS Total)	16	9	9
	Estimated Efficiency	90.7	91.1	93.2
	Shaft Horsepower	76,100	67,400	43,900
MODEL	H (FT)	1287	635	650
	Q (CFS)	10.56	20.9	18.0
	N (For Performance Test (RPM)	2390	2300	2860
	Estimated Efficiency	88.0	89.4	91.0
	Scale Factor (Prototype Model)	4.93/1	4.67/1	5.56/1

Preliminary model efficiencies provided by the model testing firms are given in Chapter 9, for the respective pump types and, in two cases, represent values obtained in sophisticated test set-ups which have been calibrated under DMJM supervision. The models are exact reduced scale reproductions of the prototype pumps for which detailed designs were prepared by each bidder. It should be mentioned here that all the efficiencies obtained on the recently conducted tests on the models were equal or higher than the predicted efficiencies.

In Volume II, Chapter 8, will be found an analysis of all available data on so-called step-up methods for calculating the prototype efficiencies from the available model efficiency. The average formula developed in this study can be used for stepping up the model efficiencies obtained, using proper corrections for balancing leakage, etc., which should bring the results of all three models to a common basis. Both this method and the method preferred by the individual test firms have been used for the predicted prototype efficiencies given in Table 9-II. It is considered more appropriate to accept the model firm method in order to account for difference in model manufacturing techniques and test methods.

4. Cavitation and Submergence

The cavitation criteria for the Tehachapi pump station pumps is minimum damage over the life of the system. However, the importance of efficiency in the economic picture means that the pump design must be optimized for efficiency rather than for minimum submergence. Then, necessary submergence to prevent cavitation damage is to be designed into the plant.

In early investigations the use of submergence values corresponding to a Suction Specific Speed of about 7900 was proposed by manufacturers and was used by the Department of Water Resources in the plant layouts.

Although this value of S is satisfactory in most kinds of applications, it is felt that final determination of S and the submergence for Tehachapi should await completion of model testing. The contracts for the model programs have called for observation windows in the model inlets so that a visual determination of cavitation can be made in addition to the usual cavitation-performance loss tests. Thus, a better knowledge of cavitation damage potential is possible.

Since some existing pumps do suffer from cavitation erosion at S values of 7900, DMJM feels that design work preliminary to completion of model cavitation tests should be more conservative and should utilize an S value of 7000. NPSH values and pump elevations based on this value are given in Table 9-I. The model tests and studies of material resistance to cavitation erosion will permit final specification of pump cavitation requirements.

For a more detailed discussion of the subject refer to Chapter 3D of Volume II.

5. Long Life, Reliability and Maintenance

a. A major requirement of the pumps for the Tehachapi Crossing will be long life and reliable operation with minimum maintenance. These characteristics are gained by specifying design parameters which assure that stress, corrosion, vibration, wear and cavitation limitations are not exceeded. For example, judicious selection of materials of construction for various pump elements will help eliminate problems associated with stress, wear and corrosion. Further, material selection will be dependent upon the quality of water being pumped at various times throughout the year.

b. Pump impeller life and maintainability can be enhanced by consideration of the following points:

(1) Generous submergence allowance in the plant design to minimize the occurence of the cavitation.

(2) Comprehensive model testing of impeller and suction elbow designs to minimize non-uniform flow patterns which would increase the occurence and severity of cavitation.

(3) Selection of corrosion and erosion resistant impeller materials for the type of water that will be pumped.

(4) Provide for a smooth impeller surface finish to reduce fluid friction and to delay the starting of wear due to corrosion, cavitation, erosion and particle erosion.

(5) Closely controlled machining tolerances at the impeller leading edge to reduce total cavitation and separation.

(6) Accurate stress analysis and conservative safety factors and working stresses.

c. Pump wear ring, interstage seal ring and balance labyrinth design should consider the following points:

(1) Provide for conservative head per length of throttling surface by use of multiple throttling surfaces and liberal lengths.

(2) Corrosion and erosion resistant material to be selected as required for the type of water to be pumped.

(3) Sufficient clearance and close tolerances to eliminate any possibility of rubbing and/or seizure.

(4) Accurate vibration analysis related to clearance and tolerances and utilizing experimentally verified radial and axial thrust data.

d. Pump casing design should likewise include proper material selection for the water that will be pumped and a thorough vibration and stress analysis. Insurance of long, trouble free bearing life can be obtained by:

(1) Selection of the best grade of available bearing material based on the extensive experience accumulated in this field.

(2) Provide for adequate lubrication during all normal and fault conditions.

(3) Provide for bearing alarm and unit trip functions.

e. A major factor in obtaining a pump in which the adverse effects of reliability and maintenance are minimized is to provide a design which may be simply and readily maintained. For example, careful measurements of wearing ring clearances at periodic intervals or possibly continuous measurements with designed-in instruments and recorders will provide maintenance personnel with data on maintenance requirements. The knowledge of specific maintenance requirements permits orderly preplanning, parts ordering and repair period scheduling.

f. Emphasis should be placed on scheduling major repairs and replacements on the basis of economics. This scheduling will occur at frequent intervals, and will consider actual wear that has taken place. The optimum time for the next maintenance action will be predicted based on the accumulated cost of reduced pump efficiency versus the cost of maintenance action. Overall plant maintenance schedules should likewise be frequently and periodically revised and updated to provide a firm schedule for immediate maintenance activities and a tentative schedule for the coming months and years.

g. Pump reliability is enhanced by designing components to operate for ling times between repairs and to provide for optimum ease of performing the maintenance actions. The following points should be carefully and thoroughly explored and analyzed to design for good maintainability:

(1) Minimize the complexity of maintenance tasks.

(2) Provide for rapid recognition of and attendance to unexpected failures or degraded performance.

(3) Eliminate or minimize design features which require frequent and/or sensitive adjustments.

(4) Insure pump and motor shaft alignment under normal and abnormal conditions by proper foundation design.

(5) Provide for good accessibility and, where practical, use sectionalized housings and rings for rapid disassembly and assembly about the pump centerline. Provide adequate inspection ports.

(6) Insure provision of all necessary tools and test equipment.

(7) Provide for fully qualified maintenance personnel.

(8) Provide for optimum unit and personnel safety.

(9) Provide for adequate service manuals.

(10) Conduct sufficient inspections to permit planning of maintenance actions on an economic basis.

(11) Consider the possibility, extent, and consequences of plant modernizations and/or revisions that will take place during the life of the plant.

(12) Plan operating schedules so as to minimize the accumulation of unit starts to the maximum extent permitted by the overall

water transportation system operations and demands, and also predict the accumulation of unit starts during the life of the plant and include this with the pump design criteria.

h. To help attain long life, DMJM has initiated studies of reliability associated with pump design and operation. The results of these reliability analyses will provide operational parameters which permit orderly and planned maintenance and will minimize unplanned outages or failures consistent with pump design requirements. By conducting a well planned program of preventative maintenance and repair, the pumping capacity will be maintained over a longer period of time.

6. Speed and Specific Speed

The initial choice of specific speed was governed by the desire to attain the maximum possible pump efficiency. The experience of the pump industry points to the range from N_s = 1800 to 3000. In view of the size of the pumps and the desire to keep submergence reasonable, a choice of 2000 was made. Pumps for the various lifts could not be exactly designed for 2000 because of the division of flow into equal increments and the specific choices of drive motor synchronous speeds. After trial computation, the speed and number of units were selected which gave specific speeds of 2160, 2040 and 1990, respectively, for the single-lift, two-lift and three-lift concepts.

The choice of the actual operating speed for the pumps was partly dependent on the specific speed choice but was primarily determined by drive motor limitations. Speeds up to 1200 RPM were considered very early in the program. The motor manufacturers felt that 720 RPM was top for the horsepower range of 75,000, and 600 RPM was considered reasonable and was chosen for the one-lift and two-lift schemes.[1] For the three-lift arrangement, the single stage pumps have a higher head than the head per stage of the other pump types, and would require greater submergence. Therefore, in order to keep the specific speed as low as possible and avoid excessive submergence, but still stay near the 2000 "optimum" N_s value, a drive speed of 514 RPM was selected.

For a more detailed discussion of the subject refer to Chapter 3B of Volume I.

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[1] See Volume II, Chapter 4 for more recent opinions of motor manufacturers on suitability of motor speeds and sizes.

CHAPTER 6

THE SINGLE-LIFT CONCEPT

A. PROTOTYPE PUMP

1. Design Conditions

In the single-lift concept, the total static lift of 1929 ft. will be accomplished at one pumping plant. With the hydraulic losses included, the design head is 1951 feet. Based on the initial assumption of 5000 cfs total delivery, the flow would be divided into 16 pump units of 312.5 cfs each.

The combination of Allis-Chalmers, Milwaukee, and Sulzer Bros., Switzerland, was chosen for the preliminary design work and model testing.

Specified operating conditions for the pump and tentative design specification for the plant and auxiliary equipment are summarized below:

a. Pump Specifications

(1) Total Dynamic Head, H — 1951 ft.

(2) Flow Rate, Q — 312.5 cfs

(3) Rotating Speed, N — 600 RPM

(4) Stages — 4

(5) Mounting — Vertical, single-flow.

(6) Submergence — To be specified - (see A.4. of this chapter).

(7) Permissible Discharge Pressure Rise — 30% of normal.

(8) Service — Water pumping, continuous duty.

b. Plant Conditions

(1) Geographical Location — Zone V, California Plane Coordinate System.

(2) Plant Elevation Approx. 1245 foot ground elevation.

(3) Pumping Pond Elevation

Normal	1239 feet
Minimum	1229 feet
Maximum	1240.5 feet

(4) Terminal Canal Elevation

Normal	3167.6 feet
Minimum	3162 feet
Maximum	3173 feet

(5) Water Inlet Passages 7-foot diameter x 100 feet long; trash rack and Bell Mouth intake.

(6) Discharge Valves Spherical Valve

(7) Motor Drive -- Electrical 13.8 kv Synchronous, 60 cycle.

(8) Water Quality To be specified later.

c. Transient Conditions

(1) Start-up Full voltage starting is planned, with pump casing full of water.

(2) Normal Shutdown Controlled valve closure.

2. Pump Design and Description

A cross-section of the prototype pump design is given in FIG. 6-1. FIG. 6-2 shows the installation requirements of the pump. It should be noted that the cross-section shows some alternative design features that presumably will be subject to final selection at a later date. The Allis-Chalmers/Sulzer description is given in the following sections. Only the format has been edited:

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

3. Pump Performance

The efficiency of the prototype pump must be predicted from past experience with model tests and full scale pumps. When the current model testing has been completed, a better estimate of prototype efficiency can be made. A-C/Sulzer has submitted the detailed procedure for their efficiency prediction which includes data on other pumps and considerable analysis of losses. The complete A-C/Sulzer analysis is given in Chapter 12. A. of Volume II. Abstracts from the A-C/Sulzer report and other performance criteria are presented in the following paragraphs:

a. Specific Speed

The initial specifications of head, flow, shaft speed and number of stages dictated the specific speed:

$$N_s = \frac{NQ^{\frac{1}{2}}}{(H_{Stage})^{\frac{3}{4}}} = 2160$$

(Q in gallons per minute)

A special consultation with all model test firms resulted in a DMJM report entitled, "Efficiency-Specific Speed Relationships," October 1964, that confirmed earlier convictions on the optimum specific speed range. The value of 2160 is in the middle of the optimum range.

b. Efficiency Prediction

Sulzer based the main part of their performance prediction on a previously completed pump and model program, the Z'Mutt pump, which is similar to the proposed Tehachapi pump and had a similar size model. The performance of the Tehachapi model was predicted from the Z'Mutt model with certain corrections for the size difference and balancing labyrinth. The size correction was based on a "step-up" formula developed by Sulzer from numerous tests on pumps and is the same as the model to prototype "step-up" formula. Having corrected the Z'Mutt model efficiency from 88.9% to 88.31% for the labyrinth, it was "stepped-up" to 88.0% for model size and speed differences. The "step-up" formula used was:

$$\frac{1-\eta'}{1-\eta} = \left[\frac{D_2}{D_2'}\right]^{\frac{1}{6.5}} \left[\frac{H}{H'}\right]^{\frac{1}{28}} \qquad (6\text{-}1)$$

Using this formula again, the prototype efficiency was predicted to be 90.74%, rounded off to 90.7%. The ratios were:

$$\frac{D_2}{D_2'} = 4.93 \text{ and } \frac{H}{H'} = 1.515$$

Sulzer presented further data, justifying the use of Equation (6-1).[1]

c. Performance Curve

A prediction of the entire performance range of the prototype pump was made by Sulzer and is presented in FIG. 6-3.

4. Cavitation Characteristics and Submergence Requirements

As can be seen in FIG. 6-3, Sulzer is recommending a NET POSITIVE SUCTION HEAD of 85 feet at the design flow rate of 3125 cfs. This corresponds to a Suction Specific Speed of:

$$S = \frac{NQ^{\frac{1}{2}}}{(NPSH)^{\frac{3}{4}}} \cong 8{,}000$$

The model testing will determine actual cavitation characteristics, but for predicting pumping plant design, the following considerations are offered. The value of S = 8000 is close to the Hydraulic Institute Recommendation corresponding to S = 7900. However, cavitation erosion of impellers is known to occur in some existing pumps with values of S this high. A more conservative value would be S = 7000, requiring NPSH = 102.5.[2]

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[1] See Chapter 8, Volume II, for an extensive analysis on "Step-up," and Chapter 12, Volume II for the Allis-Chalmers/Sulzer complete write-up.

[2] Detailed calculations are given in Chapter 3. D, Volume II.

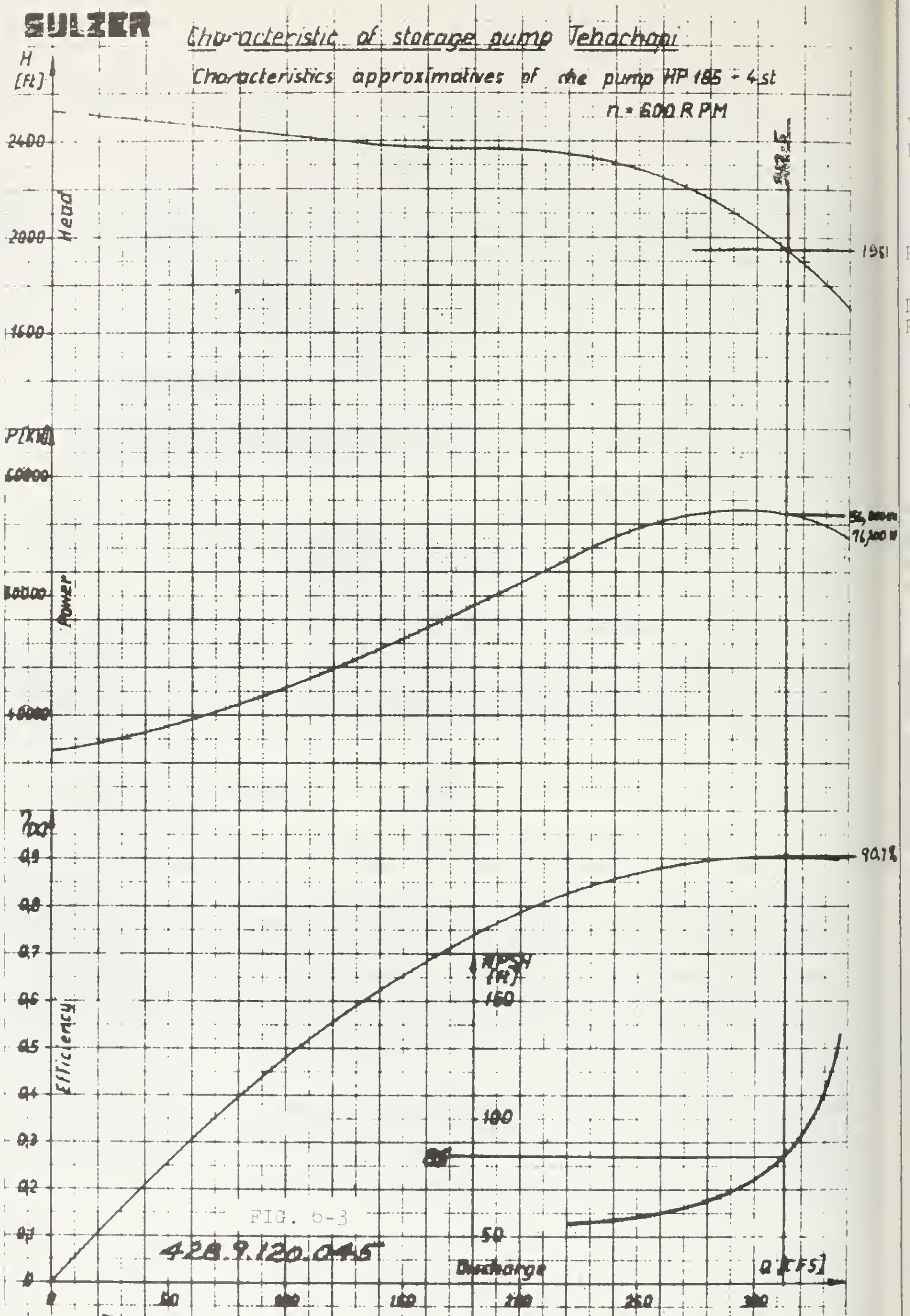

SULZER
Characteristic of storage pump Tehachapi
Characteristics approximatives of the pump HP 185 - 4 st
n = 600 RPM
H [Ft]
Head
2400
2000
1600
1951
P [KW]
Power
60000
40000
56,000
76,300
η
Efficiency
0,9
0,8
0,7
0,6
0,5
0,4
0,3
0,2
0,1
0
90.1%
NPSH [Ft]
150
100
50
Discharge
Q [CFS]
0
50
100
150
200
250
300
428.9.120.045

FIG. 6-3

With an atmospheric head of 32.5 feet at the approximate pump elevation of 1100-1200 feet, and with the average head corresponding to vapor pressure of 1.2 feet, the submergence would be:

	S	NPSH	Submergence	Elevation of Pump Inlet Stage
Per A-C/Sulzer	8,000	85.0 feet	53.7 feet	1185.3 feet
DMJM Tentative Recommendation	7,000	102.5 feet	71.2 feet	1167.8 feet

where the submergence is measured from the normal pond level of 1239 feet.

If the intake pond dropped to the minimum level of 1229 feet, then the reduction in submergence would effectively increase the Suction Specific Speed by approximately 500. Thus S would increase approximately to 8500 from 8000, and to 7500 from 7000. From this consideration, the conservative recommendation of submergence corresponding to S = 7000 (normal pond level) is made, subject to model testing where a different value of NPSH can be justified. Further, if the Tehachapi pumping system should have circumstances of operation or variation in design tolerances that would result in a flow rate higher than specified, then greater submergence would be required. Note how the NPSH curve on FIG. 6-3 rises with only a modest increase in flow rate.

It should be noted that this pump utilizes an inlet volute (see FIG. 6-1) and the flow entering the first stage impeller will undoubtedly be less uniform than with an axial approach. So, the cavitation performance of this pump will probably not match that of an axial inlet design.

5. Pump Materials

A summary of A-C/Sulzer proposed pump materials is given in Table 6-I. Wear test results and other studies may eventually result in recommendation of different materials.

Table 6-I
Tentative Selection of Materials for A-C/Sulzer Four Stage Pump

ITEM	MATERIAL		
	FROM	ASTM	DESCRIPTION
Pump Casing & Spiral	Castings Welded	A 27 Grade 70-36 (Normalized)	Carbon Steel
Top Cover		A 27 Grade 70-36	Carbon Steel
Suction Bend	Casting	A 27 Grade 70-36	Carbon Steel
Impellers	Castings (Finished All Over)	A 296 Grade CA-15 (Heat Treated)	13% Chrome Steel
Diffusers	Castings (Finished All Over)	A 296 Grade CA-15 (Heat Treated)	13% Chrome Steel
Return Channels	Casting	A 27 Grade 70-36	Carbon Steel
Balancing Labyrinth Rotating Member	Casting	A 296 Grade CA-15 (Heat Treated)	13% Chrome Steel
Balancing Labyrinth Stationary Member	Casting	B 144 Alloy 3A	Bronze
Interstage Sleeves	Cast	A 296 Grade CA-15	13% Chrome Steel
Impeller Wearing Rings	Cast or Bar	A 296 Grade CA-15	13% Chrome Steel
Casing Rings		B 144 Alloy 3A	Bronze
Interstage Bushings		B 144 Alloy 3A	Bronze
Shaft	Forging	A 235 Class E (Normalized)	Carbon Steel
Stuffing Boxes			Babbitt Lined Bronze Segments
Bearings		Not Specified	
Bearing Supports			Fabricated Steel

6. Pump Stresses and Critical Speed

A-C/Sulzer submitted a rather extensive critical speed analysis and stress analysis on the preliminary design. This analysis was reviewed and is presented in complete form in Chapter 12A of Volume II. Pertinent data and comments follow here:

a. Critical Speed

Calculations of critical speed show that the combined motor and pump critical speed will be 1297 RPM. An approximate mechanical design of the motor rotor was made for use in the critical speed calculation. The calculated critical speed is safely over the 600 RPM operating speed. The weight of impellers was found to be 8120 LB. Separate calculations for pump and motor gave:

Pump shaft alone $\approx$ 960 RPM Critical

Motor shaft alone $\approx$ 1337 RPM Critical

The motor shaft controls the combined effect. With final design of both motor and pump, critical speed should be rechecked.

b. Coupling Stress and Shrink

The approximate drive horsepower of 77, 000 HP is equivalent to a torque of 8, 080, 000 in-lb. at 600 RPM. Referring to the coupling section on FIG. 6-1, the shaft O.D. is 19.7" and the coupling section O.D. is 28.1". A shrink fit radial pressure of 9285 psi will handle the torque for a friction drive with a safety factor of two. This radial pressure can be provided by a shrink fit of .0241 in diametral difference in the shaft and coupling with .00977 in radial expansion of the coupling and .00226 radial compression of the shaft. The corresponding surface stress is 27, 200 psi.

c. Shaft Stresses

With a torque load of 8, 080, 000 in-lb. and a thrust load estimate of 340, 000 lbs. (hydraulic thrust + weight of rotating components), the principle normal stress is 6350 psi and the maximum sheer stress is 5630 psi. (Since the time A-C/Sulzer first submitted the thrust estimate and stress calculation, the balancing labyrinth was redesigned to reduce the thrust load on the thrust bearing. Presumably, this will reduce the stress above the balancing labyrinth and may reduce it to some extent below the labyrinth.)

Using impeller keys of 2-1/2" square section and 13-1/2" long, the sheer stress is 5580 psi and the compressive stress is 11,100 psi.

d. Cover Stresses

An approximate method for calculating the head cover stress ("Bach Method") gave a stress of 13,950 psi. 36 four-inch bolts will handle the cover load with a working stress of 20,000 psi.

e. Casing Stresses

At the test pressure of 1281 psi, the maximum hoop stress would be 17,600 psi, the average hoop stress (at the average diameter) would be 10,150 psi and the "average" radial displacement would be .0165 inch.

f. Stay Ring Stresses

With a simplified stress model, for the test pressure of 1280 psi, the vane bending stress = 9640 psi, the stress due to vertical forces = 8660 psi and the total is then 18,300 psi. At the working pressure of 884 psi, the bending stress = 8000 psi, stress from vertical forces = 6040 psi, and total stress = 14,040 psi.

The shell stress with the 1280 psi test pressure would be 9070 psi.

7. Cost and Weight Estimates

The price estimate received in March 1965 from Allis-Chalmers is:

16 units = $10,750,000.00 FOB Milwaukee

This corresponds to 671,875.00 each. The unit weight is 416,000 LB.

B. MODEL TESTING PROGRAM

1. Model Testing Firm:

The single-lift system model pump test program will be a joint venture of the Allis-Chalmers Mfg. Co. and Sulzer Bros. Ltd. The design, fabrication and testing of the model pump will be done by the Sulzer Bros., Ltd., Winterthur, Switzerland. The firm of Motor-Columbus, Baden, Switzerland, has been contracted by DMJM to witness all model testing and coordinate testing activities directly with DMJM.

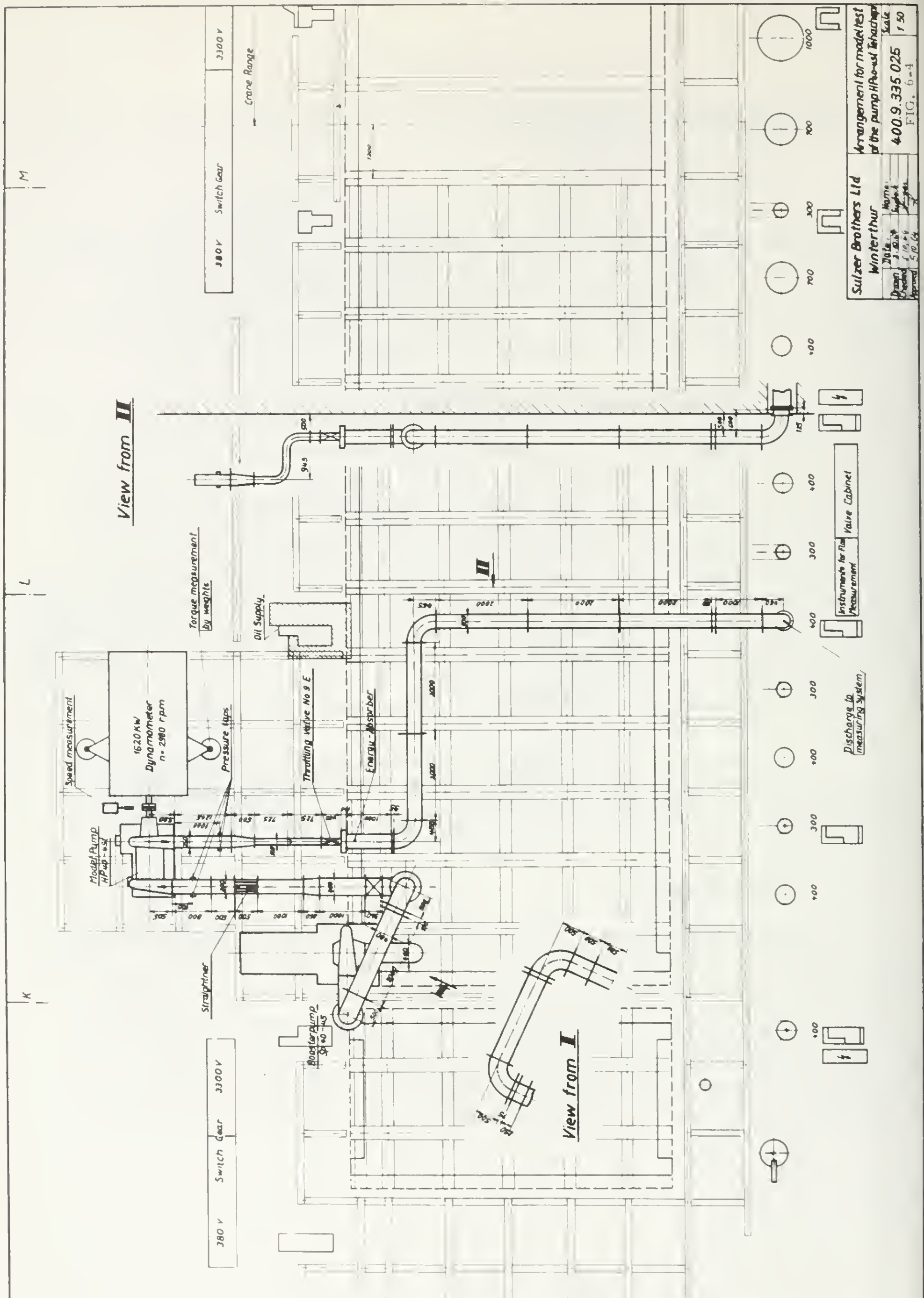
2. Description of Testing Laboratory

The Sulzer Brothers pump testing facility is contained in a building 148 x 75 feet. Located under the test floor, there is a main water reservoir with a volume of approximately 290,000 gallons. At one end of the main reservoir is another tank of approximately 43,000 gallons with a depth of 33 feet, which is used for cavitation and nozzle calibration tests. Next to the main reservoir, also underground, are four parallel sides of sufficient length for orifice or nozzle meter runs for flow measurement. The maximum measureable flow capacity is more than 100 cu. ft. sec. All valves in the pipe system can be inspected to check for leakage. The total electrical power available is approximately 10,000 KW, with voltages up to 10,000 volts. Alternating current of 60 CPS and direct current are both available.

A plan view of the laboratory arrangement for the model test is shown in Fig. 6-4 (Drawing No. 400.9.335.025). Additional drawings of the laboratory and a description of test equipment is presented in Volume II, Chapter 1.

3. Model Pump

Sulzer has fabricated parts to make two model assemblies: the four stage model of the prototype and a single stage model. The single stage model will be used for cavitation tests and general exploratory testing. A cross-section of the four stage model and the model mounting arrangement are shown in Figs. 6-5 and 6-6 (Sulzer Drawings 400.0.110.049 and 400.0.110.046). Drawings for single stage model are given in Fig. 6-7 and 6-8 (400.0.110.044 and 400.0.110.047), and the arrangement of the model on the test floor is shown in Fig. 6-9 (400.9.335.027).


View from II
View from I
Crane Range
380 V
Switch Gear
3300 V
Torque measurement by weights
Oil Supply
Speed measurement
1620 KW Dynamometer n = 2980 rpm
Pressure taps
Throttling valve No 9 E
Energy absorber
Model Pump HP 40-45
Straightner
Booster Pump
Instruments for Flow Measurement
Valve Cabinet
Discharge to measuring system
Sulzer Brothers Ltd Winterthur
Arrangement for model test of the pump HP40-45 Tehachapi
Scale 1:50
400.9.335.025
FIG. 6-4

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

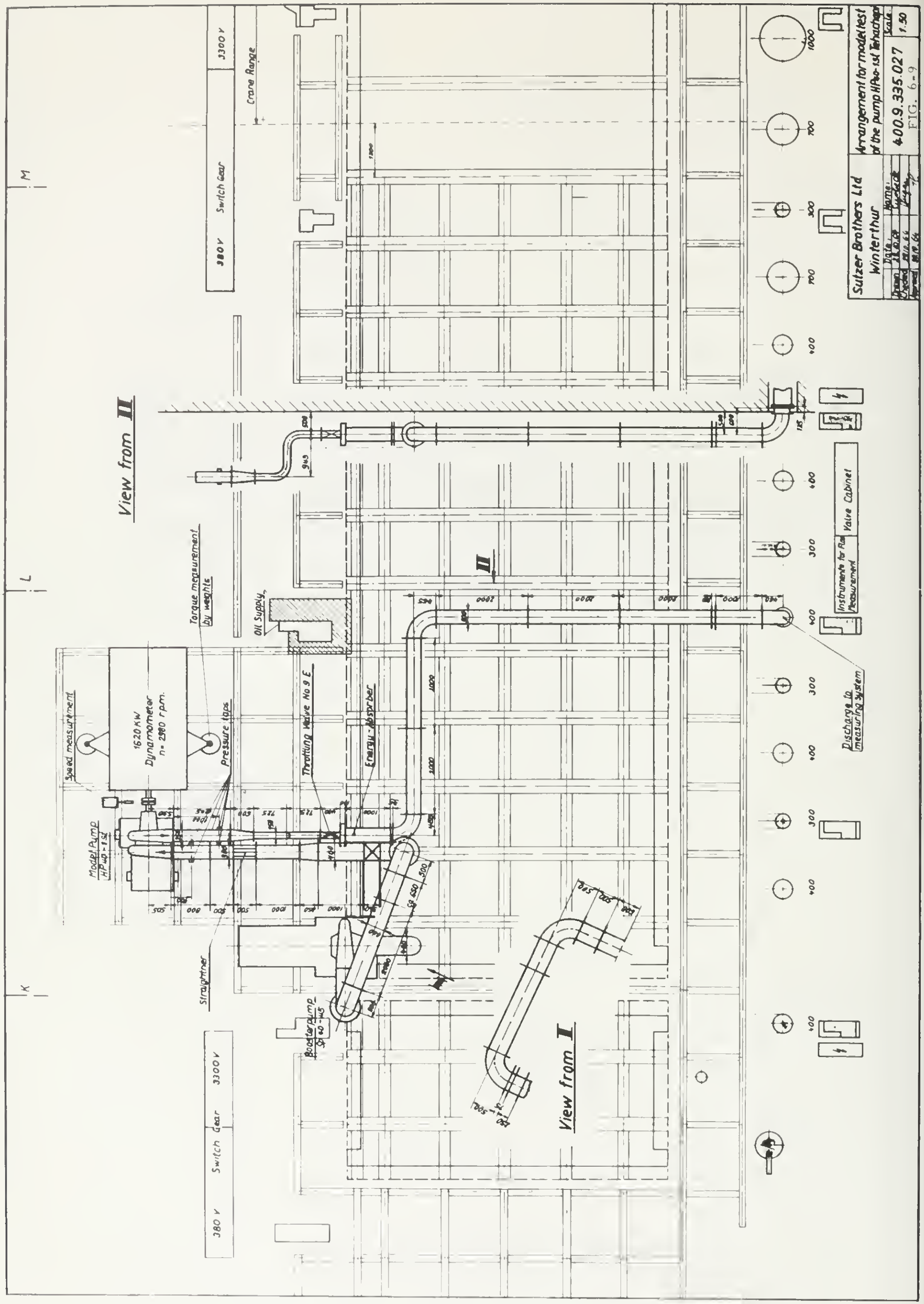

View from II
View from I
Torque measurement by weights
Oil Supply
1620 KW Dynamometer n = 2980 r.p.m.
Speed measurement
Pressure taps
Model Pump HP 40-1 st
Throttling Valve No 9 E
Energy-Absorber
Straightner
Booster Pump SP 40-45
380 V
Switch Gear
3300 V
Crane Range
Instruments for Flow Measurement
Valve Cabinet
Discharge to measuring system
Sulzer Brothers Ltd Winterthur
Arrangement for model test of the pump HP40-1st Tehachapi
400.9.335.027
Scale 1:50

FIG. 6-9

During preliminary testing, Sulzer plans to evaluate two different diffuser section designs.

4. Model Pump Test Procedures

A-C/Sulzer have submitted test procedures describing the following model pump tests:

a. Model Pump Efficiency Test
b. Cavitation Test
c. Radial Thrust Test

These procedures are reprinted in Chapter 1 of Volume II.

5. Predicted Model Pump Efficiency

Sulzer Bros. Ltd. predicts that the model pump efficiency will be 88%. A complete discussion of this point can be found in Chapter 12A, Volume II.

6. Test Schedule

The schedule of test events is being followed by DMJM through a Program Evaluation and Review Technique (PERT) Control System. Information is received from Allis-Chalmers Manufacturing Co. on the progress of each event on a bi-monthly basis. This information is processed by DMJM, injected in the PERT system for the entire program and forwarded to the Department of Water Resources. Abstract sections of the PERT chart are sent to Allis-Chalmers, advising them of the current status. A detailed description of PERT is given in Chapter 2 of Volume IV.

7. Test Results

The testing program will require several months to complete and at the time of issuing this report, only preliminary test results are available.

The preliminary tests have been made at a speed of about 1500 rpm. It has been reported simply that the hydraulic efficiency of the model is 88.1. Our consultants, Motor-Columbus, report that this is the best single test point attained. Stepped-up efficiencies using this model test value are given and discussed in Chapter 9, Volume I.

C. MOTORS

It has been assumed for this portion of the study that self-starting motors can be designed and built to reliably meet the requirements set forth below for the Motor Prototype Design. The assumption that the motors will be self-starting is based on assurances given by two of the three qualified U. S. manufacturers that the required motors are of feasible design, and notwithstanding the differences of opinion expressed by the third U. S. as well as certain European manufacturers. These differences of opinion especially with respect to the problems involved in starting the Tehachapi motors are discussed in detail in other sections of this report. The general arrangement for a typical motor is indicated on Drawing E-1--FIG. 6-10.

1. Motor Prototype Design

The 16 main pump motors are vertical synchronous type, 13,800 volt, 60 cycle, 3 phase, 1.0 power factor, rated 80,000 horsepower, 600 RPM and designed for 40% overspeed. Motors will have direct connected exciters and amortisseur windings for across-the-line starting with the pump watered and operating against a closed head. The motor will be designed for a starting frequency of one full voltage start per day.

2. Auxiliary Equipment

Motors will be enclosed in air housing with water cooling by means of air to water heat exchangers, and will be provided with high oil pressure lift on thrust bearings. Combination jacks and brakes will be included to elevate the rotor for servicing the thrust bearing and rated to brake 1% of rated torque from 50% speed to standstill in 7-1/2 minutes.

3. Motor Starting

It is assumed that the motors will be designed in accordance with American practice (as opposed to different designs which are favored by certain European manufacturers) and for full voltage starting with characteristics which will limit the starting inrush to a maximum of 250,000 KVA. This assumes that the power supply will be capable of delivering 4,000,000 short circuit KVA to the high side of the transformers which will transform transmission voltage to the required operating voltage of 13.8 KV. It is estimated that motors will accelerate from standstill to full speed in approximately 30 seconds. The starting inrush value of 250,000 KVA together with the capability of the system to deliver a minimum of 4,000,000 short circuit KVA are considered to be reasonable system design limits, which with careful scheduling of motor start up will produce system disturbances which it

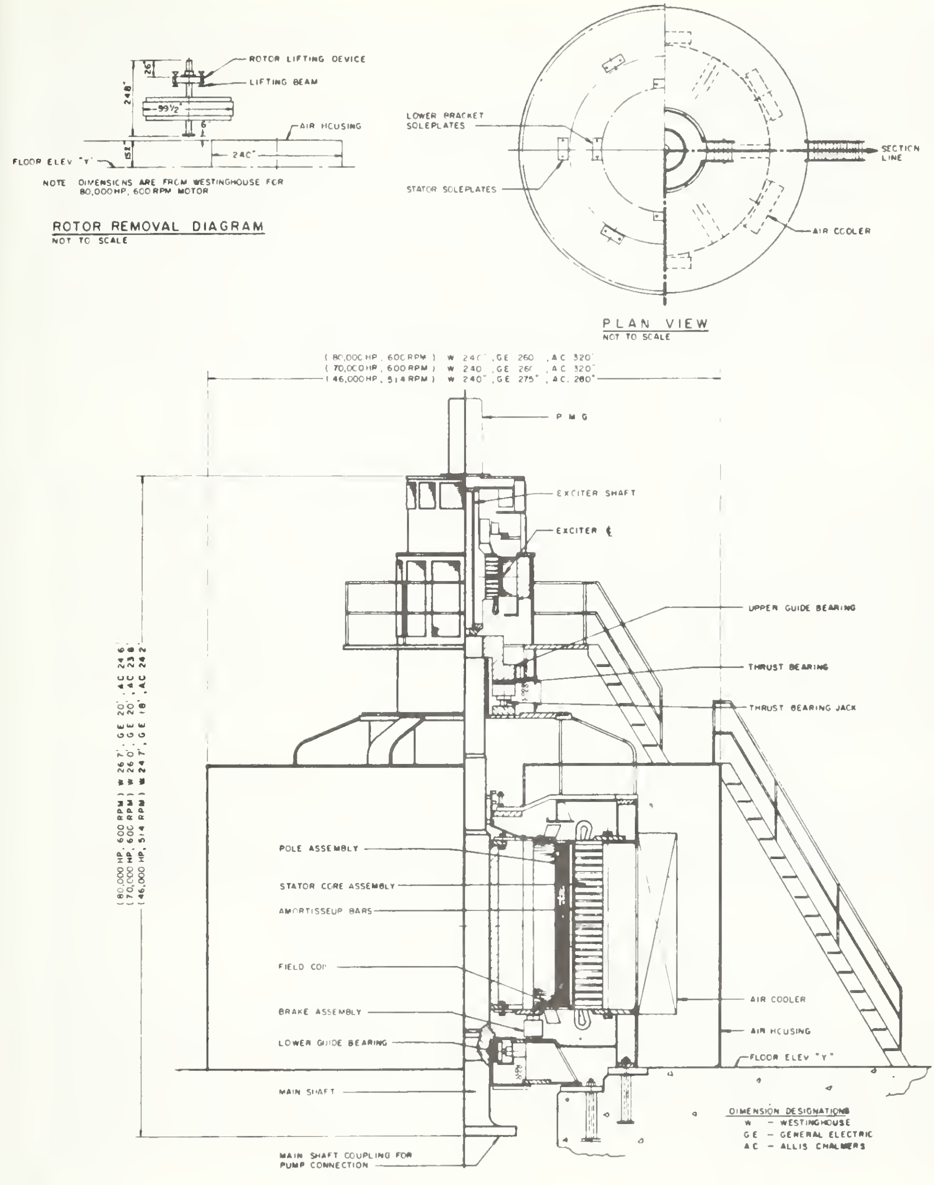


FIG. 6-10

DMJM DANIEL, MANN, JOHNSON, & MENDENHALL 3325 WILSHIRE BLVD · LOS ANGELES 5, CALIFORNIA · DUNKIRK 1-3663 PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS	TEHACHAPI PUMPING PLANT MOTOR ARRANGEMENT SINGLE-LIFT (80,000 HP, 600 RPM) TWO-LIFT (70,000 HP, 600 RPM) THREE-LIFT (46,000 HP, 514 RPM)	R BRINK W E G	637-1-1 E-1

is felt will be within permissible limits. It is therefore assumed that across-the-line full voltage starting will be used for starting the pumps during the entire life of the plant. If, however, future studies of the power system (where specifically related to the Tehachapi Pumps) indicates a need to reduce the assumed permissible inrush KVA to a value less than 250,000, it is then proposed that reactor reduced voltage starting be used. This condition is only likely to occur during the early life of the plant when starting will be most frequent due to off-peak operation and possibly coinciding with the requirement to start the motors from a power system, only partially developed to its required ultimate capacity. With growth of the power supply to match the ultimate Tehachapi load, and the change to continuous pump operation, it is further assumed that motors if required to initially start on reduced voltage will ultimately be started by across-the-line methods.

4. Power Requirements and Efficiency

Based on information obtained from motor manufacturers an efficiency value of 98.4% has been assigned to the 80,000 horsepower motors which results in a power requirement for each motor of approximately 61,000 KW with the motors delivering rated horsepower. This results in a total load of approximately 970,000 KW, exclusive of power system losses and the supporting station load requirements. For purposes of analyzing motor performance it is assumed that the 16 main pump motors will be supplied by 8 main power transformers with each transformer rated for a continuous load of 125,000 KVA and with two motors connected to each transformer.

As previously stated, it is expected that individual motors when starting across-the-line will accelerate and reach full speed in approximately 30 seconds, at which time with the discharge valve closed it is estimated that the motor will be delivering 62% of rated horsepower which corresponds to approximately 38,000 KW of electrical power input to the motor.

5. Size, Weight and Space Requirement

Major motor dimensions and total weights of the motors together with weights for the rotor and shaft assemblies are listed in Table 6-II. Dimensions as obtained from motor manufacturers will establish space requirements for the machines with allowance added to permit installation and removal of any unit and to facilitate emergency repairs and scheduled maintenance.

6. Cost Estimates

The estimated motor prices as obtained from the motor manufacturers are given in Table 6-II. These prices include the following with each motor:

Exciter including exciter cubicle

Air cooler

Air housing, platform, handrails and stairway

Thrust and guide bearings

High pressure lubricating system

Foundation bolts, half coupling and CO_2 system piping

Brakes and jacks

Current transformers for differential relaying

Space heaters

Handling equipment and unit accessories

Rail freight to destination, except where otherwise noted.

The prices given in Table 6-II do not include the cost of required 13.8 KV switchgear, power transformers station switch yard equipment or buses required to connect the motors to the control switchgear.

TABLE 6-II MOTOR DATA RECEIVED FROM MOTOR MANUFACTURERS

	Single Lift			Two Lift			Three Lift		
	West.	G. E.	A. C.	West.	G. E.	A. C.	West.	G. E.	A. C.
Horsepower	80,000	*80,000	80,000	70,000	*70,000	70,000	46,000	46,000	46,000
Speed (RPM)	600	600	600	600	600	600	514	514	514
Number of Units	16	16	16	18	18	18	27	27	27
Cost per Unit (Motors only)	$ 880,000	$ 590,000	$ 600,000	$ 704,000	$ 530,000	$ 552,778	$ 496,000	$ 460,000	$ 398,150
Total Cost (Motors only)	14,080,000	9,440,000	9,600,000	12,672,000	9,540,000	9,950,000	13,392,000	12,420,000	10,750,000
Estimated Efficiency including Excitation and Bearing Losses (%)	98.5	97.95	97.90	98.45	98.00	97.80	98.4	97.75	97.50
Inertia, Dimensions and Weights									
Motor WK^2 (Lb. Ft^2)	1,110,000	2,000,000	2,700,000	980,000	1,700,000	2,300,000	750,000	1,500,000	1,380,000
Outside Diameter over Air Housing (In.)	240	260	320	240	260	320	240	275	280
Overall Height including Exciter (Ft)	26.667	20	24.6	26.00	20	23.8	24.7	18	24.2
Rotor Diameter (In.)	99.50	101	136	99.5	101	136	101	110	118
Stator Diameter (In.)	160	157	220	160	157	220	160	170	183
Rotor Weight (Lb.)	172,000	231,000	230,000	157,000	220,000	195,000	135,000	146,000	185,000
Total Weight (Lb.)	410,000	476,000	470,000	380,000	428,000	410,000	385,000	295,000	335,000
Design Characteristics									
Synchronous Reactance Xd (%)	140	92	100	150	92	100	150	92	100
Transient Reactance X'd (%)	29	38	31	31	38	29	31	38	33
Subtransient Reactance X''d (%)	25	25	27	27	25	26	27	25	28
Motor Starting Impedance (%)	25.0	25	27	27	25	26	27	25	28
Motor Pull-In Impedance (%)	51.3	--	50	55	--	50	55	--	50
Motor Starting Power Factor (%)/Motor Pull-In P.F. (%)	7/38	--	23/53	7/38	--	23/53	7/38	--	25/53
Estimated Speed at Pull-In (% of Syn. Speed)	97	--	95	97	--	95	97	--	95
Estimated Starting Time, Standstill to Pull-In (Secs.)	30	--	15-23	30	--	15-23	30	--	15-23
Locked Rotor KVA at Rated Full Voltage (%)	400	--	370	370	--	385	370	--	350
Starting Torque at Rated Full Voltage (%)	20	--	40	20	--	40	20	--	40
Pull-In Torque at Rated Full Voltage (%)	78	--	100	70	--	100	65	--	100
Pull-In Torque at Rated Full Voltage (%)	150	150	150	150	150	150	150	150	150
Performance Data Associated with Assumed Power Supply									
KW Equivalent of Motor Rated Horsepower	60,600	60,900	61,000	53,100	53,300	53,400	35,100	35,100	35,300
Assumed Impedance of Power System (% on Motor KVA Base)	1.5	1.5	1.5	1.3	1.3	1.3	0.9	0.9	0.9
Assumed Impedance of Transformer Supplying Motor(" " " " ")	3.6	--	2.9	4.7	--	3.9	4.7	4.0	3.9
Motor Starting Inrush (KVA)	201,000	--	195,000	161,000	--	171,000	108,000	117,000	108,000
Estimated Terminal Voltage at Start (%)	81.5	82	80-90	81.7	84	80-90	82	83.7	80-90
Estimated Terminal Voltage at Pull-In(%)	91.5	--	80-90	91.5	--	80-90	92	--	80-90
Starting Torque at Estimated Terminal Voltage (%)	13	--	--	13	--	--	14	--	--
Pull - In Torque at Estimated Terminal Voltage (%)	65	--	--	58	--	--	54	--	--

*MOTORS NOT RECOMMENDED FOR SELF-STARTING BUT BY SYNCHRONOUS BACK-TO-BACK STARTING.

D. VALVES

1. Design and Operation

Three general types of valves can be considered for the single-lift concept pump discharge valve. These are the spherical valve, the needle valve, and the cone plug valve. Sufficient precedent exists for the application of any of these three valve types for the Tehachapi single-lift scheme.

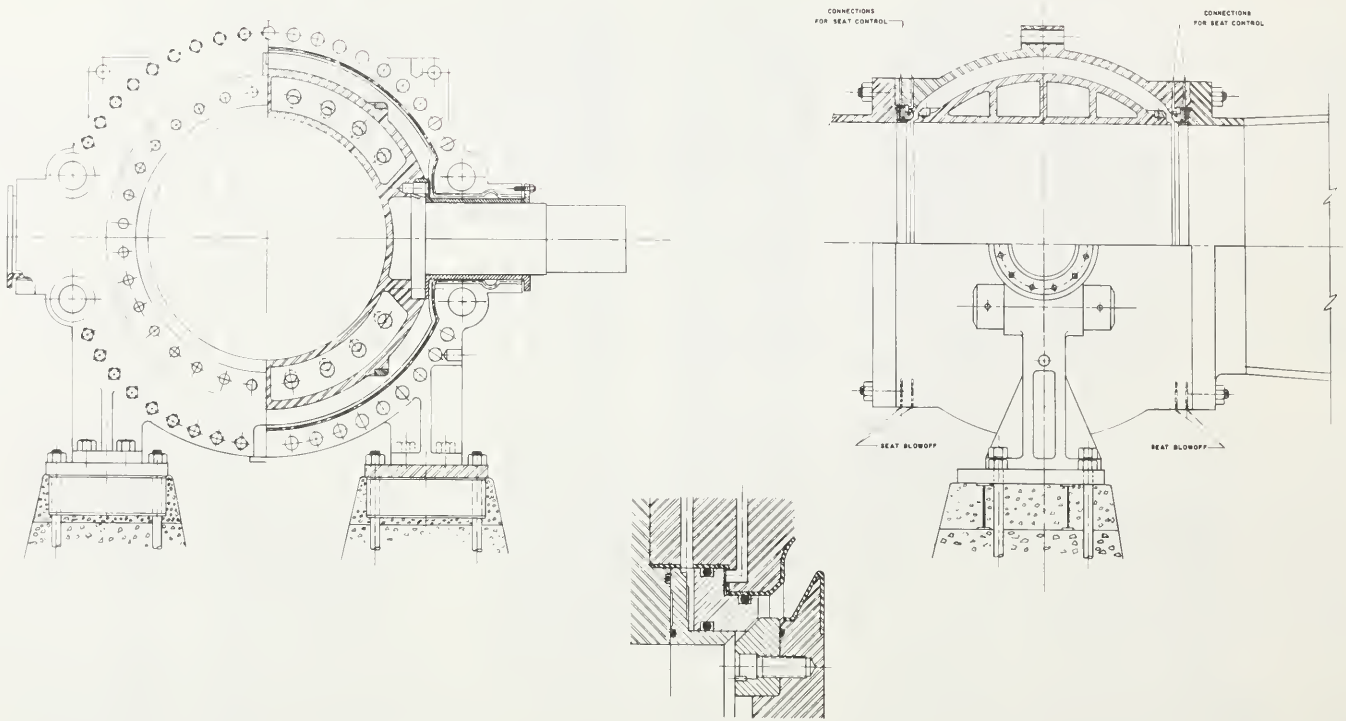
Figure 6-11 is a cross-section assembly of a typical spherical valve. In this drawing, double moveable body seats are shown. The valve is normally equipped with an operator sequencing mechanism to insure that body seats are open prior to plug rotation. An American manufacturer's only report of a field problem was due to improper operation and the sequencing devices remedied this problem.

The spherical valves can be made with double seats, one normal seat at the inlet end and one emergency seat at the outlet end. Replacement or repair of the normal seat can be accomplished without draining the penstock when the emergency seat is closed. It has, however, been pointed out by a manufacturer of cone plug valves that no seats on their cone plug valves have been replaced in more than 30 years of hydraulic service.

The cone plug valve is very much like the spherical valve in that it is a rotary valve and it affords a straight through path for water flow. Head loss in rotary valves is the smallest among all types of valves. Both types have good sealing qualities and both are simple and compact in their construction. Conical shape of the cone plug valve eases the severe fitting process of valve rotor to valve seat because of its wedging action seating properties.

The needle valve is composed of a spindle shaped interior hull inside a circular streamlined valve body. Water flow is stopped or adjusted by moving the conical part of the valve body upstream and downstream. Among its merits are good water tightness, smooth flow and little head loss though more than that experienced in the rotary valve. Its valve seats are also rarely damaged but the fact that its interior operating mechanism is perpetually immersed in water increases chances for rust or wear to disrupt its smooth operation. It is also a comparatively large and heavy valve.

In Europe, existing plants were found to be divided approximately half-and-half in the application of spherical valves and needle valves. Three



SEAT DETAIL
SEATS IN CLOSED POSITION

FIG. 6-11

DMJM DANIEL, MANN, JOHNSON, & MENDENHALL 3325 WILSHIRE BLVD · LOS ANGELES 5, CALIFORNIA DUNKIRK 1 3663 PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS	THE RESOURCES AGENCY OF CALIFORNIA Department of Water Resources	TEHACHAPI PUMPING PLANT VALVE STUDY CROSS SECTION ASSEMBLY SPHERICAL DISCHARGE VALVE	637-1-1 M-11 3-12-65

plants surveyed used both types in the same plant on different units. Out of 25 European pumped storage plants, 1 U. S. pumped storage plant (Taum Sauk), and 2 European hydroelectric plants surveyed, the following tabulation and comparison with Tehachapi requirements resulted:

Valve Type	Number of Plants	Head Range feet	Flow Range cfs	Size Range inches
Spherical	15	508 - 2064	83 - 2450	19.7 - 108
Needle	17	250 - 3151	26 - 1161	19.7 - 106
Tehachapi (Single-Lift)		1951	312.5	48.5 - 52
Tehachapi (Two-Lift)		975	555.5	63 - 66
Tehachapi (Three-Lift)		650	555.5	60 - 63

Tapered cone plug valves have been in use since 1939 at the five Metropolitan Water District (MWD) pumping plants on the Colorado River Aqueduct. These tapered cone plug valves installed at the MWD plants are used for a head range of 146 to 447 feet and a flow range of approximately 200 to 225 cfs; their inlet diameters range from 42 to 48 inches, and the outlet diameters range from 57 to 60 inches.

Modern spherical valves are entirely suited to the application and are the least expensive type. Therefore, only spherical valves are further considered for the Tehachapi crossing. For the single-lift scheme, the pump discharge valve must be capable of opening or closing with 1100 psi differential across the valve under emergency conditions. Exploratory calculations have determined that the closing cycle of the valve should be adjustable to close the first 80% of the port area in 10 to 20 seconds and the remaining 20% of port area in another 20 to 40 seconds. Seating operations involving a small fraction of the total flow may be expected to take place within approximately 15 seconds. The valve opening speed need not be adjustable, but may be expected to take place within a reasonable period of time such as 60 to 90 seconds.

Each valve is normally furnished with its own operating power package and air balasted accumulator. One manufacturer recommends that the accumulator have sufficient useable capacity to stroke the operating cylinder 3 times (3-90° rotations) in the event of oil pump failure or outage for maintenance. The operating power package normally includes motor driven oil pump, filter, oil reservoir, strainer, oil pump discharge relief valve, check valve, and oil flow control valves.

2. Size, Weight, and Costs

The size of the pump discharge valve should be no smaller than the pump discharge connection. Forty-eight and one-half inches (48.5") is the size advanced by the model manufacturer for the single-lift concept. The spherical valve will be from 6 to 8 feet long, depending on the flange arrangement at the ends. An adaptor between the pump discharge and the discharge valve will be approximately 7 feet long, or can be longer. The valve will weigh about 150, 000 lbs., but can vary depending on the selected configuration and end connections on the valve.

The valve operator will occupy a position adjacent to the valve, where the torque shaft extends through the valve body. The hydraulic cylinder should extend about two or three feet past the valve body, resulting in an overall width of 10 to 12 feet and a length of 6 to 8 feet. The overall height from floor to top of valve may be 12 or more feet, depending primarily on the connection of the operating mechanism, whether on the floor, a wall, pier, or on the valve body itself. The cost of a 48.5 inch spherical valve with operator and with the features recommended will be approximately \$160, 000. A 52 inch valve will cost approximately \$180, 000. An additional \$40, 000 might be charged for development costs.

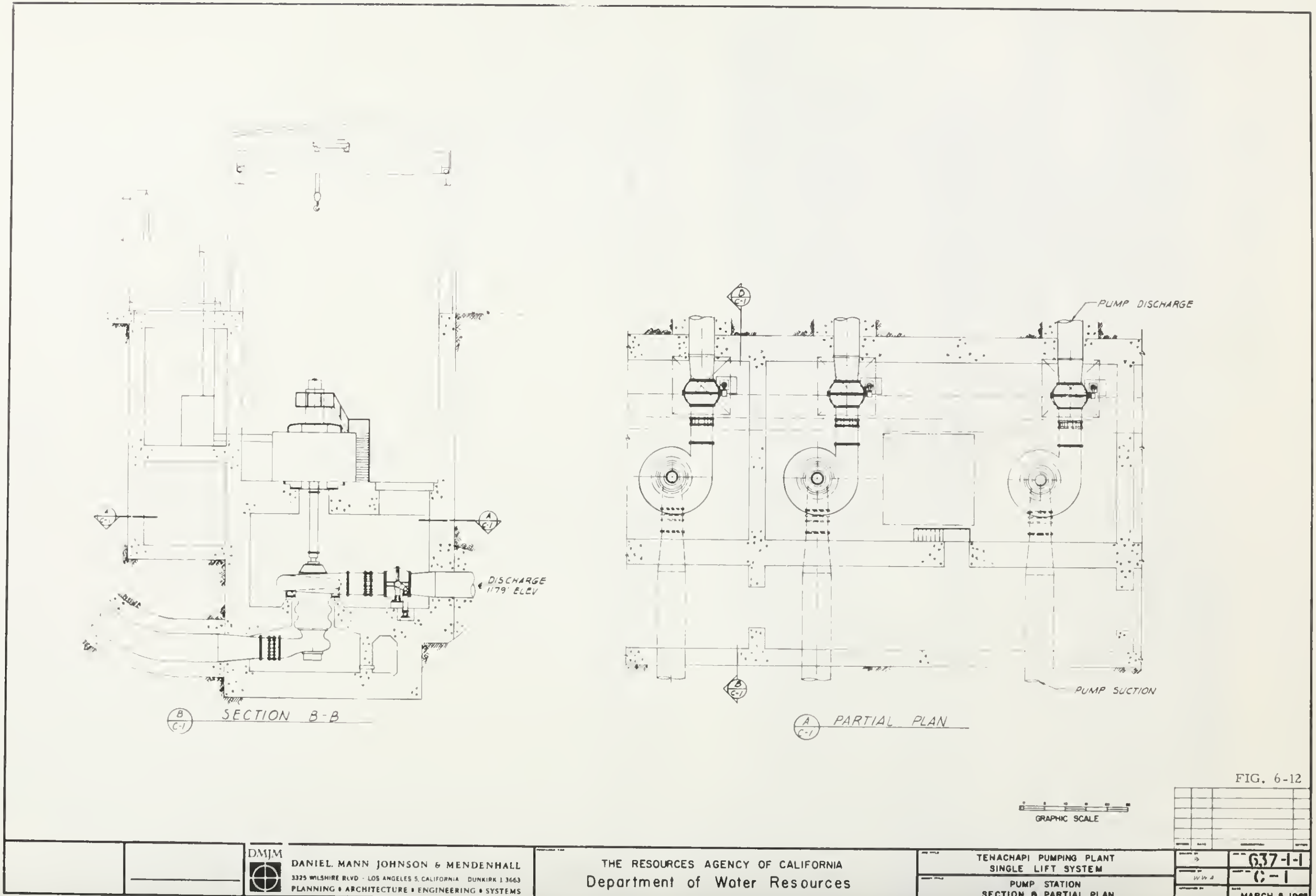
E. PLANT ARRANGEMENT

FIG. 6-12 is a cross-section drawing of a plant showing pump, motor and valve in the general arrangement suited to the single-lift concept. This is by no means a final recommendation, as numerous details must be considered in the plant layout, once the lift concept has been chosen.

The pump is supported under the discharge volute and can be completely removed from its mount for overhaul. The motor arrangement is typical of vertical installation with an intermediate shaft connecting the motor and pump. The motor and pump can be aligned on their mounts, and then the tie-bolt couplings tightened to make pump, valve and embedded piping an integral structural unit.

The four-stage pump has a balancing ring for hydraulic thrust control and normally operates with approximately 57 tons total load (hydraulic thrust + weight of rotating parts). At start-up, the load may reach 90 tons, so a thrust bearing in the motor of 100-ton capacity will be needed. Final design will require a check on weights of all rotating parts, as the intermediate shaft length may vary some and the motor rotor weight will be known with greater accuracy.

The estimate of the critical speed made by Allis-Chalmers for the unit is 1297 RPM. This figure must also be recalculated when all details of motor and intermediate shaft are known.


SECTION B-B
DISCHARGE 1179' ELEV
PUMP DISCHARGE
PUMP SUCTION
PARTIAL PLAN
GRAPHIC SCALE
DMJM
DANIEL, MANN, JOHNSON & MENDENHALL
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PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS
THE RESOURCES AGENCY OF CALIFORNIA
Department of Water Resources
TEHACHAPI PUMPING PLANT
SINGLE LIFT SYSTEM
PUMP STATION
SECTION & PARTIAL PLAN
637-1-1
C-1
MARCH 5, 1965

FIG. 6-12

F. OPERATION AND MAINTENANCE

1. System Elements

Elements of particular interest with regard to operation and maintenance are the pumps, motors, valves, switchgear and penstocks. In this preliminary report, stress has been placed on those areas which offer greatest contrast when comparing the lift concepts. Elements which differ most markedly from one concept to another are the pumps, motors, and valves.

The one lift concept is notable for its shaft seal, thrust balancing, starting, and high pressure discharge problems. An important offsetting feature, however, is the pump, motor, and valve layout which features sixteen units operating in parallel. The two and three lift concepts will have nine units operating in parallel in each plant. Since the one lift plant will have a greater number of parallel units, repair and maintenance down time provide an availability that offers inherent reliability advantages over concepts featuring units which must operate in series.

2. Precedent

Precedent established in installed systems throughout the world for the one lift concept's single flow, multistage pumps as planned at Tehachapi is more limited than that for the two lift concept's double suction pumps. Experience with both the one lift and two lift concept pumps surpass that for the three lift. Information from twelve plants featuring single suction multistage pumps provides a major portion of the basis for this report. (see Chapter 2, Volume II). Three of these twelve pumps have vertical posture and only one required power comparable to the 75,500 HP planned for Tehachapi (58,000 HP at Lunersee). However, eight other plants surveyed had motors exceeding 50,000 horsepower and twenty in all used the vertical arrangement. The number of stages used in the single suction, multistage pumps surveyed ranged from two through nine with one being a four stage pump. Heads per stage varied from a low of about 270 feet to a high of 770 feet with a majority of them being in excess of the 500 foot Tehachapi prototype head.

Such basic elements of the reliability and maintenance study as wear ring life are based on information from all pumps surveyed, interstage seal ring lives on all multistage pumps, and balance ring lives on all single suction pumps.

3. Operation

Operation of each of the lift concepts should be quite similar except for detailed techniques of plant start-up and shut-down. Intermittent operation based on demand and power availability will probably be the initial mode. This mode is to be succeeded by virtually continuous operation in later years. Predicted yearly demand buildup and resultant accumulated operating hours are given in Tables 6-III and 6-IV. Here, the differences in pump utilization through the years illustrate the available down time for repair and maintenance.

Although this preliminary study has concentrated most heavily on comparative aspects for continuous operation, a subject of much interest has been the effects which number of starts for each unit has on maintenance and reliability. Some preliminary findings may be presented on this subject.

Efforts were made in the reliability study (Chapter 7, Volume II) to determine the effect of number of starts on wear of pump components. A record was made of average number of starts per thousand operating hours for each plant surveyed. This record was compared with times to repair of wear rings, balance rings, and impellers. The times were weighted for material, water velocity and percentage of clearance increase. Only plants having similar water quality were used to compare successful operating times and starts. Table 6-V presents number of starts and times between repairs in thousands of hours.

A study of the data in Table 6-V reveals no pattern of decreased life with increased starts. Results contradictory to the expected trend are much in evidence. No definite correlation could be made between unit starts and component life for pumps.

An area of controversy exists regarding unit starts and times between repair for motors. One manufacturer bases his guarantee on the number of unit starts because a start puts increased stress on end turns and increased heat loads on amortisseur windings. Two other manufacturers believe the motor may be designed to take almost any frequency of starts desired without paying a penalty in increased maintenance. Field records on motor problems reveal no definite correlation. However, many plants practicing frequent starts connect their motor to the pump after it is brought up to speed by a turbine or other device.

It is suggested that thrust bearings suffer practically no wear with the possible exception of that which may occur at the instant of starting or stopping when the oil film is thin. Wear is small even under these conditions

YEAR	YEARLY DEMAND BUILDUP		Required Offpeak Pumping Capacity,	ONE-LIFT			
	Acre-Feet, In Thousands	Corresponding Continuous Flow Rate CFS	CFS	Number of Installed Units	Installed Capacity, CFS	AVERAGE Yearly Operating Hrs./Unit	Utilization Factor % of Total Year
1971	200.0	276	600	4	1250.0	1950	22.1
1972	414.2	572	1230	4	1250.0	4000	45.7
1973	530.9	733	1576	6	1875.0	3400	39.1
1974	647.2	894	1922	7	2187.5	3600	40.9
1975	763.2	1054	2266	8	2500.0	3700	42.2
1976	1,056.4	1459	3102	10	3125.0	4100	46.7
1977	1,346.8	1860	3929	13	4062.5	4000	45.8
1978	1,638.1	2263	4758	16	5000.0	3950	45.2
1979	1,929.6	2665		"	"	4650	53.2
1980	2,221.2	3068		"	"	5400	61.4
1981	2,506.2	3462		"	"	6050	69.2
1982	2,791.2	3855		"	"	6750	77.2
1983	2,908.7	4018		"	"	7050	80.4
1984	3,026.2	4180		"	"	7300	83.6
1985	3,144.5	4343		"	"	7600	86.9
1986	3,262.7	4508		"	"	7900	90.2
1987	3,328.2	4597		"	"	8050	91.9
1988	3,393.6	4688		"	"	8200	93.8
1989	3,419.9	4724		"	"	8300	94.5
1990	3,446.1	4760		"	"	8350	95.2
1991	3,451.2	4767		"	"	"	95.3
2000	"	"		"	"	"	"
2010	"	"		"	"	"	"
2040	"	"		"	"	"	"

TABLE 6-III. ASSUMED YEARLY DEMAND BUILDUP
(YEARLY OPERATION ROUNDED TO NEAREST 50 HOURS)

YEAR	UNITS IN OPER.	HOURS PER YEAR	ACCUMULATED OPERATING HOURS						
			Units 1-4	5-6	7	8	9-10	11-13	14-16
1971	1-4	1950	1,950						
1972	1-4	4000	5,950						
1973	1-6	3400	9,350	3,400					
1974	1-7	3600	12,950	7,000	3,600				
1975	1-8	3700	16,650	10,700	7,300	3,700			
1976	1-10	4100	20,750	14,800	11,400	7,800	4,100		
1977	1-13	4000	24,750	18,800	15,400	11,800	8,100	4,000	
1978	1-16	3950	28,700	22,750	19,350	15,750	12,050	7,950	3,950
1979	"	4650	33,350	27,400	24,000	20,400	16,700	12,600	8,600
1980	"	5400	38,750	32,800	29,400	25,800	22,100	18,000	14,000
1981	"	6050	44,800	38,850	35,450	31,850	28,150	24,050	20,050
1982	"	6750	51,550	45,600	42,200	38,600	34,900	30,700	26,800
1983	"	7050	58,600	52,650	49,250	45,650	41,950	37,750	33,850
1984	"	7300	65,900	59,950	56,550	52,950	49,250	45,050	41,150
1985	"	7600	73,500	67,550	64,150	60,550	56,850	52,750	48,750
1986	"	7900	81,400	75,450	72,050	68,450	64,750	60,650	56,650
1987	"	8050	89,450	83,500	80,100	76,500	72,800	68,700	64,700
1988	"	8200	97,650	91,700	88,300	84,700	81,000	76,900	72,900
1989	"	8300	105,950	100,000	96,600	93,000	89,300	85,200	81,200
1990	"	8350	114,300	108,350	104,950	101,350	97,650	93,550	89,550
1991	"	"	122,650	116,700	113,300	109,700	106,000	101,900	97,900
1995	"	"	156,050	150,100	146,700	143,100	139,400	135,300	131,300
2000	"	"	197,800	191,850	188,450	184,850	181,150	177,050	173,050
2005	"	"	239,550	233,600	230,200	226,600	222,900	218,800	214,800
2010	"	"	281,300	275,350	271,950	268,350	264,650	260,550	256,550
2020	"	"	364,800	358,850	355,450	351,850	348,150	344,050	340,050
2030	"	"	448,300	442,350	438,950	435,350	431,650	427,550	423,550
2040	"	"	531,800	525,850	522,450	518,850	515,150	511,050	507,050

TABLE 6-IV. ACCUMULATED OPERATING HOURS
ONE - LIFT

Water Quality	Starts per 1000 hrs.	TIMES BETWEEN REPAIRS, HRS. x 10^{-3}		
		Wear Rings	Balance Rings	Impellers
Good	2	30, 56, 62, 89		103
Good	100	3		
Good	300	101		10, 11
Poor	40	5		
Poor	60	14		
Average	2	22		
Average	20	79, 23	18	
Average	260	35		
Average	400	36		
Average	500		19	

TABLE 6-V. UNIT STARTS AND PUMP COMPONENT LIVES FOR SURVEYED PLANTS

because of an adsorbed coating of lubricant on the bearing surfaces. It may be that the amount of bearing wear which occurs on start and stop will not be significant over the years of intermittent operation at Tehachapi. If this wear is considered a problem, it presumably may be remedied through the use of high pressure lubrication systems. Again, no proportionate increase in bearing maintenance with number of starts may be seen in field data.

A logical presumption is that wear in the seats and packing of a valve is related to the number of times the valve is operated. The same may be said for breakers and switchgear. In fact, one plant makes a practice of refurbishing its breakers after every forty starts. Repairs of these items generally are so few and far between that the data allows no reasonable comparisons between time to repair and number of starts.

To determine the increased costs of designing for or maintaining a system employing ten or twenty starts per day rather than one, and to contrast these presumably increased costs with increased benefits involved, a study of much greater scope would be necessitated. On the basis of investigations to date it can only be stated that no statistically fool proof method is available which could associate pump wear and frequency of starts within the limits of common pumping practice. It appears that many plant operators tend to exaggerate the effects which unit starts have on pump component life.

One point which was forcefully brought out by the reliability study was the importance of keeping records. Every operating plant could be treated as a laboratory performing research in problems of design, operation and wear if uniformly good records prevailed throughout. Accumulated pump hours for each pump should be metered and placed with the dates of maintenance actions. Accumulated unit starts should also be listed. All ring clearance checks should be kept in the records, particularly initial and final clearances. Clearances should be checked at a minimum of four points on the rings. Ring leakage flow in cfs at the time of and immediately after each maintenance action should also be recorded when possible. Leakage flows should be given for fixed bushing stuffing boxes if clearances can not be accurately determined. In addition, records of repair times and descriptions should be maintained. Repair costs would also be helpful and are required if optimum times for repair based on economics are to be determined.

Most important of all records which should be kept but are usually neglected are those regarding water quality. Periodic samples should be taken. Water pH should be monitored and the sample should be weighed and filtered. The filtered residue is weighted to get weight of undissolved solids per unit weight of water. If possible, several grades of porosity should be used in filtering to determine average size of undissolved solids. Acid tests might be used to distinguish limestone content from quartz or sand. If records such as these were kept at every plant, effects of water quality on wear would soon become apparent.

4. Maintenance

Maintenance of the single lift concept is distinctive mainly because of the required thrust balancing devices and the size of the units. Thrust balancing labyrinths generally are replaced more often than wear rings so that the four-stage prototype pump should require a greater number of scheduled maintenance actions than the other prototypes. The fact that four stages must be maintained makes pump overhaul repair time longer for the single lift than the three-lift.

Throughout the text of this interim report, scheduled or planned maintenance is considered to be only that which is necessitated by predictable wear of pump components. In reality, this constitutes only a small part of pumping plant maintenance (possibly ten percent of total outage time). Other types of outages normally encountered in practice include preventive maintenance of motors, valves, bearings, switchgear, hydraulic auxiliaries, cooling systems, penstocks, canals, and other equipment. Down time must also be allowed for inspection, testing, cleaning, and modification as a part of preventive maintenance and modernization. How much time might normally be devoted to outages other than repair of worn pump components will emerge from studies of scheduled and unscheduled repair time distributions now in progress.

Preliminary prediction of a pump maintenance schedule for the Tehachapi single lift has the fixed metal bushing shaft packing being replaced every 14,000 pumping hours. The replacement should take about 50 hours on a three-shift basis. Balance labyrinths will require overhaul every 21,000 hours and take about 100 hours to perform. Replacement of wear rings and interstage seal rings, together with repair or replacement of impellers, would take place every 31,500 hours and require 250 hours of maintenance time. For these estimates, it was assumed that balance labyrinths, wear rings, and interstage seal rings are made of 1020 or 1040 carbon steel. Forthcoming results from the wear test program will be used to determine pump operating time between repairs for other material selections. The predicted repair times and times between repair resulted from the reliability analysis presented in Volume II of this report.

The pump shaft packing is assumed to be of the fixed metal bushing type which is normally used at pump speeds similar to those planned for Tehachapi. It could, however, be a carbon ring type of fixed bushing or a self-adjusting carbon seal (mechanical seal) or even an adjustable soft packing. Of these types, the fixed metal bushing probably takes the most time per maintenance action, but it should not require repair as frequently as the other types if the metal is more resistant to wear than carbon or soft packing. Table 6-VI gives maintenance information on the various types of shaft packing in common use on large pumps.

The single lift prototype pump has a fixed bushing at either end and a four throttling surface labyrinth added to the lower bushing. Simultaneous replacement of both bushings would probably entail use of a larger crew than would be needed for the singular bushing on the three lift prototype.

Motors comparable in horsepower to those planned for the Tehachapi single lift represent advances in the field and experience with them is too limited to draw definite conclusions as to their maintainability. Motor manufacturing representatives are generally agreed that horsepower rating of the motors for the three-lift concepts should not influence choice of concept in spite of the fact that stresses ordinarily increase with horsepower. Forces increase as the square of current and deflection varies as the cube of end turn extension length. Evidently, the duty of starting any of the Tehachapi prototypes involves a similar degree of punishment to the appropriate motor.

Time when a motor is ready for overhaul is not easily determined because it is not dependent on predictable wear such as that which sets the pump overhaul time. This study regards motor maintenance as unscheduled outage time precipitated usually by cooling failures or insulation deterioration. These fatigue-type failures are probably related to operating hours and number of starts, but proper design is a compensating factor. The coils of the 65,000 horsepower motors at Grand Coulee were scheduled for stator rewinding after twenty years of operation, which is equivalent to about 35,000 operating hours per unit with probably forty or fifty starts per unit in that time. Rewinding costs with new coils are estimated at 25-33% of initial cost, and installation time is about one month. Estimates for the 75,000 HP Tehachapi stator rewinding are 17% of total machine cost and approximately six weeks maintenance time with a crew of five.

Bearing maintenance time is also assumed to be a part of unscheduled maintenance time because the greatest cause of failures is improper lubrication, excess heat due to high spots or cooling failures, or installation of inferior wearing-quality babbitt. One manufacturer claims 360

TYPE OF PACKING	MEAN TIME BETWEEN REPAIRS (operating hours)	MEAN REPAIR TIME (working Hours)	COMMENTS
soft pack	2700	8	Source of many forced outages, particularly at higher speeds.
fixed carbon bushing	8700	28	Subject to seizure.
carbon - mechanical seal	unknown*	unknown*	Unpredictable behavior, has not been proven in use. *A carbon-mechanical seal has operated at Vianden for up to 3600 hours without failure. This device is of such a contemporary nature that times to repair and of repair are not available.
fixed metal bushing	13700	47	

TABLE 6-VI. COMPARATIVE MAINTENANCE OF SHAFT PACKINGS

thrust bearings have operated more than 3000 machine years with average outage time of six hours per 50 machine years. Thrust bearing performance at Hoover Dam shows one group to have a length of service equivalent to 123 years of operation and about 12,000 starts with no failures.

Maintenance of the spherical valves used for the single lift concept is normally a routine matter of inspection, lubrication and replacement of worn packing. Items to be inspected regularly include the hydraulic control equipment, in which cleanliness and proper oil level are important, and the valve seats, in which wear or foreign matter may be found. The valve seat normally used in operation may be inspected without draining the penstock due to the pressure of an emergency seat. Valve operator fittings at all pin joints on the cylinder and torque arm assembly must be lubricated. Packing replacement for valves does not represent a significant part of overall plant maintenance and may be quickly accomplished, particularly when it is contained in a cartridge.

5. Reliability

Successful delivery of annual water requirements at Tehachapi is more compatible with the single lift concept than with either of the others. In order to deliver water, a plant must be both able to operate and available to operate. Since these two independent conditions must both be satisfied for successful water delivery, the likelihood of success is the product of ability and availability which turns out to be 0.979 for a year of single lift operation. The single-lift index (0.979) may be compared with that of the two-lift (0.950) and the three-lift (0.902) to see that a single-lift offers the best prospects of successful water delivery. Derivations of these indices are given in the report on Reliability in Chapter 7 of Volume II.

Another way to describe the probability of success is to state that the probability of the single-lift doing its job successfully each year are about 50 out of 51. Chances for successful operation are 10 out of 11 for the three-lift and 20 out of 21 for the two-lift. Reasons for the decreased chances on the two and three lifts are given in the Reliability sections of 7.F and 8.F, and Volume II. These contingencies do not take into account any restrictions on successful operation which upstream plant outages may impose on Tehachapi nor are they properly weighted for motor and valve life versus operational stresses for each lift concept. However, preliminary indications are that any inherent reliability advantages enjoyed by individual motors and valves in the two and three lift concepts will suffer greater degradation when viewed from a serious operation aspect than did the individual pump reliability advantages which have been studied in detail. The reason for this is the greater unpredictability of motor and valve outages.

CHAPTER 7

THE TWO-LIFT SYSTEM

A. PROTOTYPE PUMPS

1. Design Conditions

In the two-lift concept, two pumping plants will provide the static lift of 1,929 feet with pumps of equal total dynamic head (static head + penstock losses and discharge kinitic head) of 975 feet. Each plant would have nine (9) pumps rated at 555.6 cfs to produce an assumed total of 5,000 cfs.

The combined firms of Baldwin-Lima-Hamilton, Philadelphia, and J. M. Voith, Heidenheim, Germany, were selected to do the preliminary design and model testing.

Specified operating conditions for the pump and tentative design specifications for the plant auxiliary equipment are summarized below:

a. Pump Specifications

(1) Total Dynamic Head, H — 975 feet

(2) Flow Rate, Q — 555.6 cfs

(3) Rotating Speed, N — 600 RPM

(4) Stages — 2 (double-flow)

(5) Mounting Vertical, Double Flow

(6) Submergence To be specified see A.4 of this chapter.

(7) Permissible Discharge Pressure — 25 percent of normal

(8) Service Water pumping, Continuous Duty.

b. Plant Conditions

(1) Geographical Location Zone V - California - Plane coordinate system.

(2) Plant Elevations

Plant 1 -- Approx. 1,245 ft. ground level elevation
Plant 2 -- Approx. 2,200 ft. elevation

(3) Pumping Pond Elevation - Plant 1

Normal - 1,239 ft.
Minimum - 1,229 ft.
Maximum - 1,240.5 ft.

(4) Terminal Canal Elevation - Plant 2

Normal - 3,167.6 ft.
Minimum - 3,162 ft.
Maximum - 3,173 ft.

(5) Water Inlet Passages - 8-1/2 ft. diameter x 30 ft. long divided into two 5.25 ft. pump inlet passages (double flow).

(6) Discharge Valves - Spherical Valve

(7) Motor Drive - Electrical - 13 8 KV Synchronous - 60 cycle

(8) Water Quality - to be specified later.

c. Transient Conditions

(1) Start-up - Full voltage starting is planned with pump casing full of water.

(2) Normal Shutdown - Controlled valve closure.

2. Pump Design and Description

A cross-section of the prototype pump design is given in Figure 7-1. The installation is shown in Figure 7-2. Although the complete inlet is not shown as part of the pump, the bifurcation to the two pump inlets should be symmetrical rather than as shown in Figure 7-2. A discussion of the prototype design submitted by B-L-H/Voith follows. Only the format has been edited.

"a. Design basis for the Selection of the Prototype (Dwg. 2.21 - 19857){Figure 7-1}

For the design of the pump, two aspects have been considered:

(1) High efficiency

(2) Dependable operation

In order to attain a high efficiency, the following steps have been taken under the hydraulic angle:

The assumed velocity of the inflow to the suction branch = 16.7 fps, is comparatively low. The suction branches have circular cross-sectional areas so that a satisfactory transition to the suction pipe can readily be provided. The water will be accelerated to about 34.5 fps at the pump inlet. In the suction bend, ribs are provided which prevent pre-rotation of the water.

The distance of the impeller outlet from the guide vane inlet is large enough to minimize pressure variations at the vane tips and inside the spiral case. Also, the noise level will be low. On the other hand, the distance is not so large that a reduction of the efficiency need be expected.

The guide vanes of the first-stage radially extend over a comparatively large distance so that the flow velocity after the impeller is transformed into pressure to the largest extent possible. The pressure conversion by the return vanes is less than 5%. The transition from the outlet of the guide vanes of the first stage to the return vanes occurs in a vane-free space. As a result, no eddies can be set up. We regard this design as more suitable than extending the guide vanes up to the return vanes, one reason being that the passages can more conveniently be finished, the other reason being that in the curving vane passages no rotary components can be created.

The water flow leaving the return vanes of the first stage is accelerated slightly another time before entering the double-flow impeller of the second stage.

Regarding the transition from the impeller outlet to the inlet of the guide vanes of the second stage, the same remarks apply as for the transition of the first stage.

The radial extension of the guide vanes of the second stage is substantially greater than that of the first stage.

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

"b. Spiral Case Considerations

For the selection of the length and the thickness of the guide vanes of the second stage, design considerations have been exclusively decisive. The guide vanes in the spiral case zone are subjected to pulling forces and to deflection. The stresses stay within permissible limits. As construction material, cast steel will be used.

Regarding the installation and removal of the impeller, it should be noted that the impeller of the lower first stage is installed or removed from underneath and the impeller of the upper first stage as well as the impeller of the second stage are installed or removed from above.

"c. Balance and Seal Design

The pump is completely symmetrical so that no significant hydraulic thrust need be feared in the axial direction.

The shaft is sealed off by a sliding ring type seal. As clean pressure water is channeled into the space between the two sliding rings, a liquid film is created between the pressure surfaces so that these surfaces are not subjected to mechanical wear, while the sealing surfaces are effectively cooled. A small constant amount of water permanently leaks past the seal into the pump suction bend, thus preventing the ingress of dirty water, which in this way is kept from the sliding ring type seals. Hence, these sliding ring type seals will operate satisfactorily even with dirty water.

The pump impeller is rigidly connected to the motor. The clearance is staggered. The radial clearances of the impeller are 0.0236 ± 0.002 in., the radial clearances of the throttling bushes between the first and second stages would be 0.0177 ± 0.002 in.

"d. Pump Mounting

The pump has been so designed that the spiral case may be embodied in concrete or mounted as exposed equipment. In the latter case, the pump could, for dismantling, be shifted sideways and dismantled in a separate pit with the help of the machine-house crane.

"e. Critical Speed

The shaft has been so dimensioned that according to our experience, the critical speed will be about 25% above the highest speed in the reverse direction. A final computation cannot be made until the motor dimensions have been made available.

"f. Discharge

The delivery branch of the pump will have a diameter of 5 ft. 3 in. This gives an outlet velocity of 25.6 fps, a value which by no means is excessive. We are of the opinion that it might be possible to increase even the water velocity at the outlet of the spiral case to 32.8 or 34.5 fps with the specified delivery heads. On the other hand, with higher velocities there may be a slight loss in efficiency. In view of this, we have preferred to use the lower outlet velocity."

3. Pump Performance

The prediction of efficiency for the prototype was accomplished by estimating the efficiency of the model and then "stepping up" to the prototype size and speed. The complete write-up of the procedure, as submitted by B-L-H/Voith, is given in Chapter 12-B of Volume II. Essential data is given here:

a. Specific Speed

The specified N, Q, and H dictated the specific speed:

$$N_S = \frac{N\,Q^{1/2\,*}}{H_{Stage}^{\;3/4}} = 2050$$

In a separate letter, B-L-H/Voith has confirmed this value for N_S as the best selection to provide high pump efficiency.

b. Efficiency Prediction

Starting with a prepared efficiency chart for a single stage, single-suction pump, B-L-H/Voith made corrections for: (1) first stage return vanes; (2) shaft through the inlet; (3) reduction of disk friction, due to double-flow construction of second stage; (4) change in Reynold's number (from a graph); (5) change in clearance loss; and (6) second stage guide vanes. The resultant calculation gave η_i = 89.4% and B-L-H/Voith guaranteed 89.0% for the model.

Voith presented two formulas for step-up, one by Ackeret and one by Pfleiderer (see Chapter 12, Volume II and also Chapter 8, Volume II). Presumably, they employed the Ackeret formula:

*As this is a double-flow pump, the value of Q used in this equation is one-half the total pump Q.

$$\frac{1-\eta_i \text{ Proto}}{1-\eta_i \text{ Model}} = \frac{1}{2} + \frac{1}{2}\left(\frac{Re_{Model}}{Re_{Proto}}\right)^{0.2} - \qquad (7.1)$$

and obtained a value for the prototype of η_i = 91.0%. It was suggested that the efficiency might go as high as 91.8%.

c. Performance Curve

An estimate of the prototype performance characteristic was prepared by Voith and is presented in Figure 7-3.

4. Cavitation Characteristics and Submergence Requirements

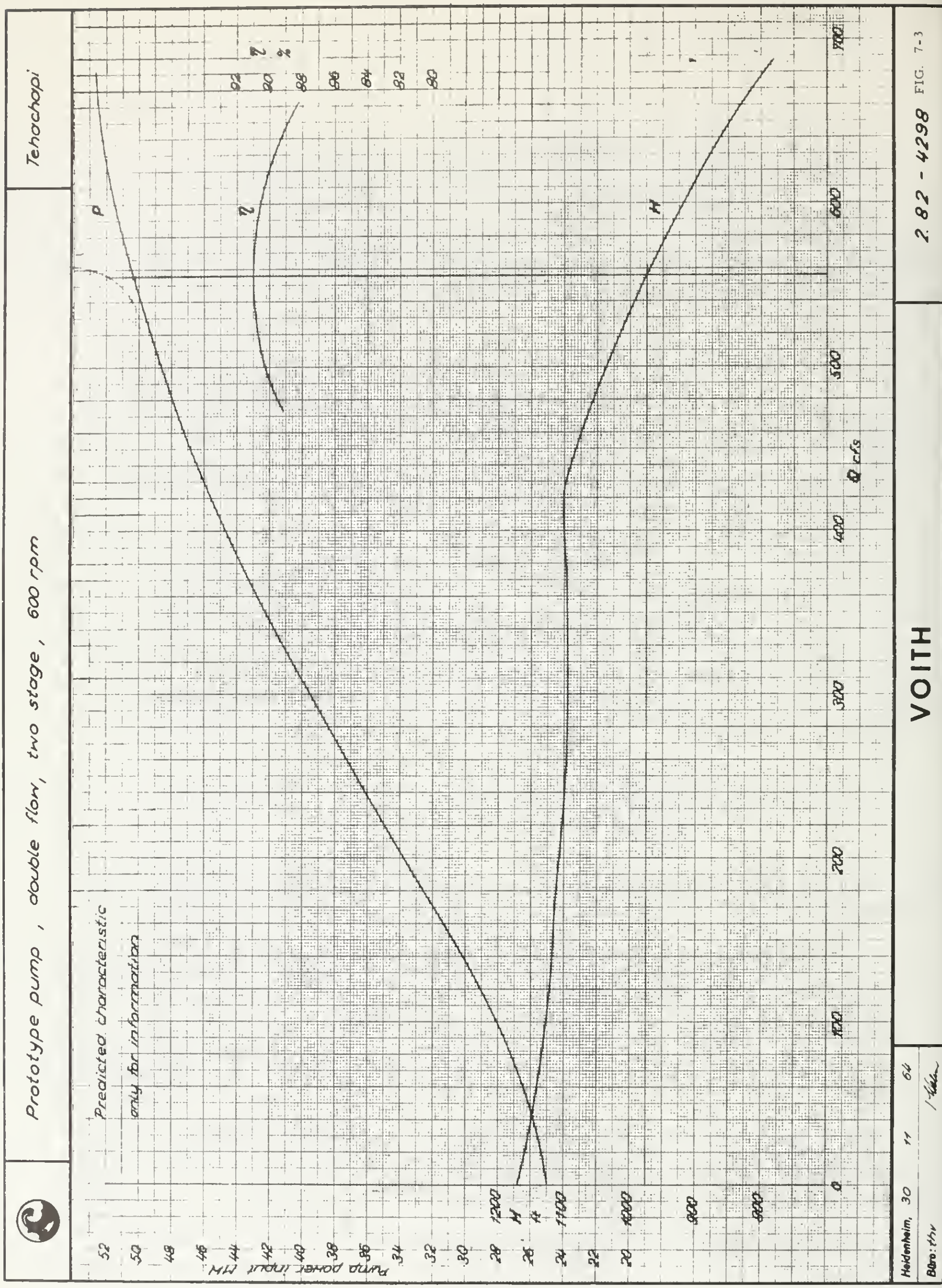
The exact submergence requirements will be determined by model testing. However, prior to tests, the recommended submergence as shown on the installation drawing, Figure 7-2, is 51 feet, measured from the water surface to the centerline of the pump discharge. Since this is a two-stage pump, this distance does not correspond to the submergence with respect to a single-stage pump. The actual submergence to the uppermost suction impeller would be about 4 feet less -- Z = 47 feet. Then, NPSH = Z + (ha-hv) = 47 +31.3 = 78.3. The corresponding Suction Specific Speed is:

$$S = \frac{NQ^{1/2}}{(NPSH)^{3/4}} = 8000$$

The recommended submergence thus effectively corresponds to the same recommendation as for the single-lift scheme, utilizing a 4-stage pump.

To be more conservative concerning cavitation damage, a value of S = 7000 should be considered.

	S	NPSH		Submergence	Approximate Elevation of Uppermost Inlet Stage
Per B-L-H/ Voith	8000	78.3 ft.	(Plant 1	47 ft.	1192 ft.
			(Plant 2	48.2 ft.	2154.8 ft.
DMJM Tentative recommendation	7000	94.5 ft.	(Plant 1	63.2 ft.	1175.8 ft.
			(Plant 2	64.4 ft.	2138.6 ft.



Tehachapi
Prototype pump , double flow, two stage , 600 rpm
Predicted characteristic
only for information
Pump power input MW
P
η
H
Q cfs
VOITH
2.82 - 4298
Heidenheim, 30 11 64
Büro: thv

FIG. 7-3

The submergence at Plant 2 is greater by 1.2 feet due to the increase in atmospheric pressure at the higher plant elevation. The normal intake water level altitude of Plant 2 is assumed to be 2,203 ft.

This pump style employs a volute type inlet and is more likely to have non-uniform flow which would increase cavitation susceptibility.

5. Prototype Mechanical Design and Materials

As part of the complete model testing contract, the model test firm is required to submit prototype design details such as stress calculations, materials, etc. This information is required in the model test firm's final report. B-L-H/Voith has not submitted such details in preliminary form, except for general information already presented in Section A.2. of this chapter.

6. Cost and Weight Estimates

In March 1965, a price of $760,000 per pump was given. For 18 units, the total pump cost would be $13,680,000. No estimate of unit weight has been submitted.

B. MODEL TESTING PROGRAM

1. Model Testing Firm

The model pump test program, as part of the two-lift system, will be a joint venture of the Baldwin-Lima-Hamilton Corporation, Industrial Equipment Division, and J. M. Voith, GMBH, Heidenheim (Brenz), Germany. The model pumps will be designed, fabricated and tested at the J. M. Voith plant.

Motor-Columbus, Baden, Switzerland, will witness all model testing and coordinate testing activities directly with DMJM.

2. Description of Testing Laboratory

FIG. 7-4, Voith Drawing 2 83-8498, shows the test arrangement for conducting the model pump tests Supply water delivery heads can be varied by means of a special pressure regulating system designed by Voith The water required for the model tests will be made available in the following way:

a. The water can be supplied directly from a high level tank. This tank is located on top of a mountain in the neighborhood of the Hydraulic Laboratory and has a diameter of about 118 feet and a height of about 26 feet. Its capacity is about 282,000 cubic feet. The geodetic difference between the highest water level in the high-level tank and the water level in the hydraulic laboratory is about 328 feet. The high-level tank is connected with the hydraulic laboratory through two pipes which can pass a water flow up to 21.2 cfs. The high-level tank is filled by storage pumps which draw the required water from a pure water spring situated close to the Hydraulic Laboratory.

b. The water can be directed to the model pump via a pressure reducer which consists of a combination of a Francis turbine and a centrifugal pump on a single shaft. The Francis turbine is supplied with water from the high-level tank, while the centrifugal pump draws its water from the above-mentioned spring. The water leaving the Francis turbine and the water delivered by the centrifugal pump is jointly channeled to the model pump. This pressure reducer is designed for a maximum water flow of 21.2 cfs. The pressure can be varied between 0 and 164 feet by means of a simple guide vane adjustment of the Francis turbine. Details of the pressure reducer are given in Volume II, Chapter 1, along with descriptions of the other laboratory equipment.

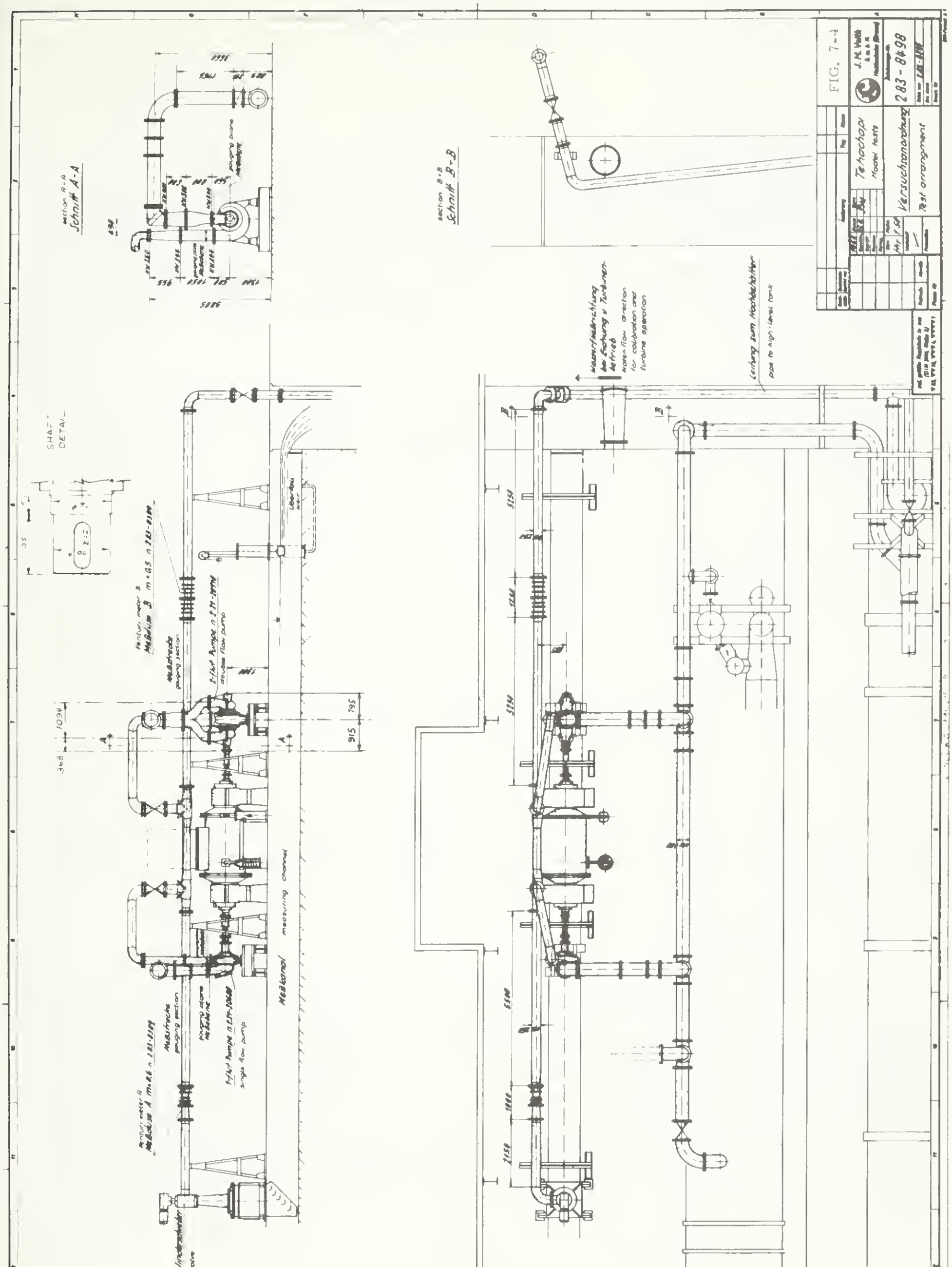


FIG. 7-4
J. M. Voith
Tehachapi
Model tests
Versuchsanordnung
Test arrangement
283-8498
section A-A
Schnitt A-A
section B-B
Schnitt B-B
Meßkanal
measuring channel
Leitung zum Hochbehälter
pipe to high-level tank
throttling valve
single flow pump
double flow pump
gauging section

3. Model Pump

J. M. Voith has planned to test both a full two-stage, double flow model and a single stage model. The single-stage model is primarily for cavitation testing. Because the impeller for an inlet stage has half the total pump capacity, the single-stage model has a "half-size" volute section. Cross-sectional drawings of the full model and the single-stage model are shown in Figs. 7-5 and 7-6. (Voith drawings No. s 2. 21-20740 and 2. 21-20680).

The inlet section is considered to be an important factor in model testing and model firms are required to model the inlet. The double flow pump requires a bifurcation and the dimensional details are shown in Fig. 7-7 (Voith drawing No. 2. 83-8541).

During preliminary testing Voith will test three different impeller designs for best performance and intends to make inlet modifications for improved cavitation performance.

4. Test Procedures

The test set-up and test procedures have been documented by Voith and are presented in Volume II, Chapter 1.

5. Predicted Model Pump Efficiency

J. M. Voith, GMBH, predicts that model pump expected efficiency will be 89.4% and that the guaranteed efficiency will be 89.0%. A complete discussion of this point can be found in Chapter 12, Volume II.

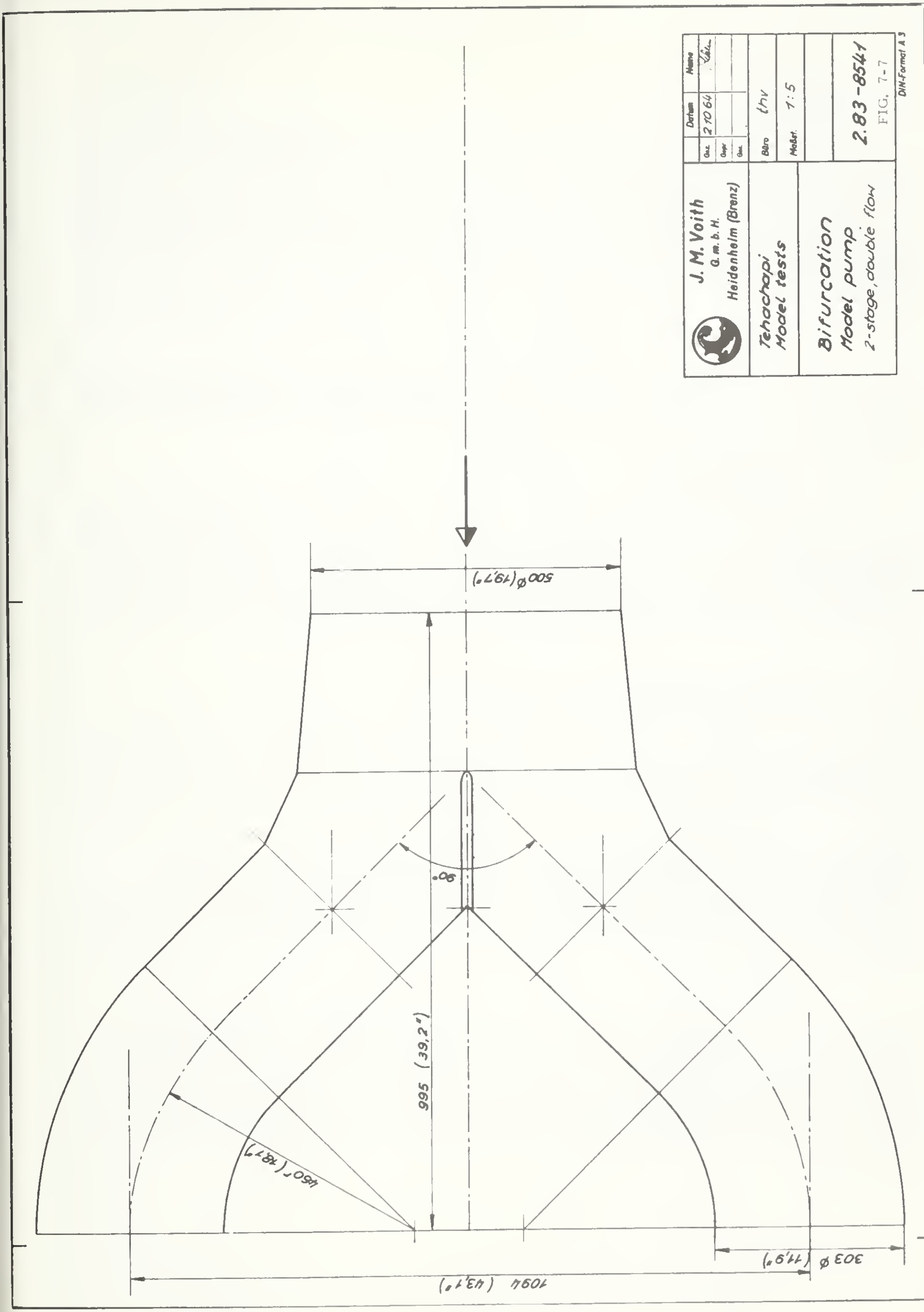
6. Test Schedule

The schedule of test events is being followed by DMJM through a Program Evaluation and Review Technique (PERT) control system. Information received from Baldwin-Lima-Hamilton Corporation on the progress of each event on a bi-monthly basis. This information is processed by DMJM, injected in the PERT system for the entire program and forwarded to the Department of Water Resources. Abstract sections of the PERT chart are sent to Baldwin-Lima-Hamilton Corporation, advising them of the current status. A detailed description of PERT is given in Chapter 2 of Volume IV.

7. <u>Test Results</u>

Preliminary test results have been received from Voith and the best of their three impeller designs has achieved an efficiency of 90.6 "Stepped-up" prototype efficiency, given in Chapter 9 (Volume I.)

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.


J. M. Voith
G. m. b. H.
Heidenheim (Brenz)
Datum
Name
27 10 64
Büro
thv
Maßst.
1:5
Tehachapi
Model tests
Bifurcation
Model pump
2-stage, double flow
2.83-8541
FIG. 7-7
DIN-Format A 3
500 ø (19,7")
90°
995 (39,2")
460 r (18,1")
303 ø (11,9")
1094 (43,1")

C. MOTORS

Except as noted below, the motors for the two-lift system are essentially the same as the motors for the single-lift system as set forth in Chapter 6C. It is also assumed that the two-lift motors will be self-starting and can be built to reliably meet the requirements set forth below for the Motor Prototype Design. The general arrangement for a typical motor is indicated on Drawing E-1 (Figure 6-10, Chapter 6C.).

1. Motor Prototype Design

The nine (9) main pump motors at each of the two plants are 70,000 horsepower, 600 RPM, 13,800 volt, 1.0 power factor vertical synchronous type and designed for 40% overspeed.

2. Motor Starting

It is assumed that motors will be started at full voltage by the same method as used for the single-lift 80,000 horsepower motors. Motors will be designed to limit the starting inrush to a maximum of 200,000 KVA. As a result, the power system disturbance resulting from the starting inrush will be somewhat less than for the single-lift arrangement.

3. Power Requirements and Efficiency

An efficiency value of 98.4% has been assigned to the 70,000 horsepower motors which is the same value used for the 80,000 horsepower motors of the single-lift system. This will result in a power requirement for each motor of approximately 53,000 KW with the motor delivering rated horsepower, or approximately 477,000 KW at each of the two stations, exclusive of power system losses and the supporting station load requirements. For purposes of analyzing motor performance, it is assumed that the nine (9) main pump motors at each of the two stations will be supplied by three main power transformers of 165,000 KVA capacity with three motors connected to each transformer.

With a starting time of 30 seconds and 55% of rated horsepower required at rated speed with a closed discharge valve, the average rate of power increase during the motor starting period will be approximately 970 KW per second, which at the end of the 30-second accelerating period will equal 29,000 KW. This will be followed by the gradual opening of the discharge valve to bring the pump and motor up to a steady state rated load condition.

D. VALVES

1. Design and Operation

The two-lift concept at Tehachapi would subject the discharge valve to approximately 975 feet of head and it would handle about 556 cfs water flow. Discussions of design, operating characteristics and applications of valves suitable to the two-lift discharge function are presented in Chapter 6, Section D and in the Technical Volume. These valve types are the spherical, the cone and the needle valves.

Information on closing times for discharge valves is given in Section 6. D. Operators are also mentioned there.

The pump inlet function for the two-lift concept might be performed by a butterfly valve since this valve is generally easier to mate with cylindrical draft tubes than is a sluice valve. Design pressure for the valve would be 50 psi nominal for approximately 64 feet of inlet head or submergence at the upper impeller of the double suction pump. Manufacturers recommend the use of metal seats since they are less susceptible to major damage from debris and since one replacement of a rubber seat should pay for the additional first cost of a metal seated valve.

The butterfly valve has a disc which is usually either slightly elliptical or perfectly round in shape and is supported on the valve body by means of a valve shaft which penetrates the center of the disc. When fully closed, the valve disc will come into a tight contact with the valve body at a position at right angles to the center line of the pump or be slightly inclined toward the center.

Butterfly valves inspected at plants surveyed for this study include a 108-inch diameter valve at Tracy with a 197-foot head and a flow of 850 cfs. The Buchanan pumping plant uses an 84-inch size for 120 feet of head and a flow of 835 cfs. At Flatiron, a 76-inch valve operates with 240 feet of head and a 360 cfs flow.

2. Size, Weight and Costs

Based on a 63-inch inside diameter spherical valve for pump shutoff, a floor space of 7 to 9 feet along the valve axis by 11 to 12 feet across the valve and operator is required. The overall height from floor to top of valve may be 12 or more ft., depending primarily on the connection of the operating mechanism whether on the floor, a wall, pier, or on the valve body itself. The valve will weigh about 170,000 pounds, but can vary depending on the selected configuration and flanged connections on the valve. The inlet butterfly valve will be approximately 3 feet long and 13 feet wide including operator. The butterfly valve will weigh approximately 40,000 pounds.

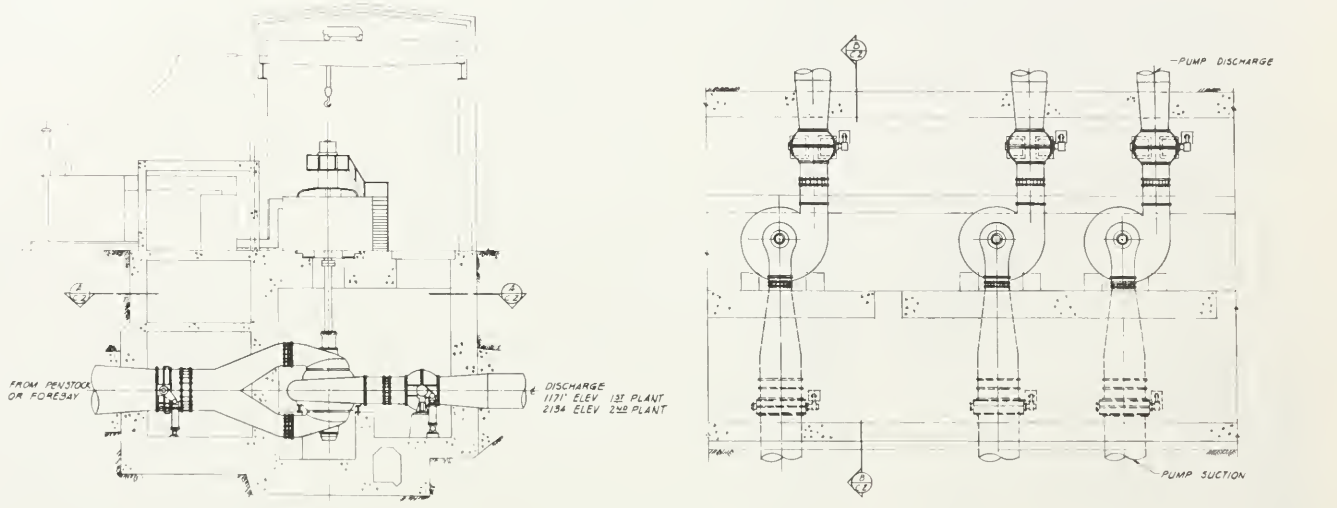
The cost of a spherical valve, 63-inch inside diameter and 550 psi design, is approximately $180,000 with operator and hydraulic supply unit. The 8-foot - 6-inch butterfly valve with operator will cost approximately $60,000.

E. PLANT ARRANGEMENT

The installation of the pump motor and valves is shown in Figure 7-8. This figure shows a cross-section of what would be Plant No. 2, where a butterfly valve on the pump inlet would be required to isolate the pump during periods of in-operation. Plant No. 1 would be similar except the inlet butterfly valve would not be present.

The pump and motor mounting are typical of a vertical installation, utilizing an intermediate drive shaft. The two-stage, double-flow pump may be completely removed from its mounting. The pump is supported under the scroll discharge. The double-flow inlet uses a symmetrical bifurcation in order to insure balanced hydraulic performance. Submergence is measured from the top inlet stage which will be more susceptible to cavitation.

The double-flow pump is automatically hydraulically balanced so that the weight of rotating components represents the total thrust bearing load at all conditions of operation. Information on the bearing load has not yet become available.



B C-2 SECTION B-B

A C-2 PARTIAL PLAN

GRAPHIC SCALE

FIG. 7-8

DMJM DANIEL MANN JOHNSON & MENDENHALL
3325 WILSHIRE BLVD · LOS ANGELES 5, CALIFORNIA · DUNKIRK 1 3663
PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS

THE RESOURCES AGENCY OF CALIFORNIA
Department of Water Resources

TEHACHAPI PUMPING PLANT
TWO-LIFT SYSTEM
UPPER PUMP STATION
SECTION & PARTIAL PLAN

WWA

637-1-1

C-2

MARCH 5, 1965

F. OPERATION AND MAINTENANCE

1. System Elements

Elements and components of unique importance for operation and maintenance of the two-lift system are its double suction input, its multistage makeup, and its series pumping mode of operation. Double suction eliminates the balance labyrinth problems of the single-lift concept because this design provides the pump inherent balance. Multistage configuration necessitates added repair time over the three-lift concept, since more components are subject to repair. Series operation results in lower reliability than the single-lift concept and higher reliability than the three-lift concept because outages in either lift affect the other, and available total repair time must be divided among an increased number of outages. Motor power and valve pressure are intermediate between the one- and three-lifts.

2. Precedent

The two-stage, double suction pump appears in more large European pumping plants than any other type. Twelve of the 29 plants visited there featured these pumps and four more had double suction single-stage units (see Chapter 2 of Volume II and Volume III). Four plants feature double suction pumps in vertical positions. There appears to be no precedent for two plants operated in series using these pumps, but the Hausern-Witznau-Waldshut complex uses two-stage, single-suction pumps in series with forebays at each plant. Operating head for most of the two-stage pumps surveyed is comparable to that planned for Tehachapi. Horsepower on four of the two-stage pumps exceeded the Tehachapi figure. In fact, power ranged from 83,000 HP to 93,000 HP on these pumps.

3. Operation

Operation of the two-lift concept involves interdependency of two plants. When a unit in one plant ceases to operate, a unit in the other plant must be shut down because of insufficient storage capacity between the plants. The fact that the two plants cannot operate independently decreases their overall reliability. Flow control is also diminished because only nine units act in parallel rather than 16 as in the single lift. Operational control is complicated by communication of a forced outage and decision-making in regard to which units should be shut down during an outage and which repairs could best be delayed until repair is also needed at the adjoining lift.

Yearly operational flow demand and accumulated hours for the two- and three-lift concepts are given in Tables 7-I and 7-II.

YEAR	YEARLY DEMAND BUILDUP		Required Offpeak Pumping Capacity, CFS	TWO AND THREE EQUAL LIFTS			
	Acre-Feet, In Thousands	Corresponding Continuous Flow Rate CFS		Number of Installed Units/Sta.	Installed Capacity, CFS	Average Yearly Operating Hrs./Unit	Utilization Factor % of Total Year
1971	200.0	276	600	3	1666.7	1450	16.6
1972	414.2	572	1230	3	1666.7	3000	34.3
1973	530.9	733	1576	3	1666.7	3850	44.0
1974	647.2	894	1922	4	2222.2	3500	40.2
1975	763.2	1054	2266	5	2777.8	3350	38.0
1976	1,056.4	1459	3102	6	3333.3	3850	43.8
1977	1,346.8	1860	3929	8	4444.4	3650	41.8
1978	1,638.1	2263	4758	9	5000.	3950	45.2
1979	1,929.6	2665		"	"	4650	53.2
1980	2,221.2	3068		"	"	5400	61.4
1981	2,506.2	3462		"	"	6050	69.2
1982	2,791.2	3855		"	"	6750	77.2
1983	2,908.7	4018		"	"	7050	80.4
1984	3,026.2	4180		"	"	7300	83.6
1985	3,144.5	4343		"	"	7600	86.9
1986	3,262.7	4508		"	"	7900	90.2
1987	3,328.2	4597		"	"	8050	91.9
1988	3,393.6	4688		"	"	8200	93.8
1989	3,419.9	4724		"	"	8300	94.5
1990	3,446.1	4760		"	"	8350	95.2
1991	3,451.2	4767		"	"	"	95.3
2000	"	"		"	"	"	"
2010	"	"		"	"	"	"
2020	"	"		"	"	"	"
2040	"	"		"	"	"	"

TABLE 7-I. ASSUMED YEARLY DEMAND BUILDUP

(YEARLY OPERATION ROUNDED TO NEAREST 50 HOURS)

YEAR	Units in Operation	Hours Per Year	ACCUMULATED OPERATING HOURS					
			Units 1-3	4	5	6	7-8	9
1971	1-3	1450	1,450					
1972	1-3	3000	4,450					
1973	1-3	3850	8,300					
1974	1-4	3500	11,800	3,500				
1975	1-5	3350	15,150	6,850	3,350			
1976	1-6	3850	19,000	10,700	7,200	3,850		
1977	1-8	3650	22,650	14,350	10,850	7,500	3,650	
1978	1-9	3950	26,600	18,300	14,800	11,450	7,600	3,950
1979	"	4650	31,250	22,950	19,450	16,100	12,250	8,650
1980	"	5400	36,650	28,350	24,850	21,500	17,650	14,000
1981	"	6050	42,700	34,400	30,900	27,550	23,700	20,050
1982	"	6750	49,450	41,150	37,650	34,300	30,450	26,800
1983	"	7050	56,500	48,200	44,700	41,350	37,500	33,850
1984	"	7300	63,800	55,500	52,000	48,650	44,800	41,150
1985	"	7600	71,400	63,100	59,600	56,250	52,400	48,750
1986	"	7900	79,300	71,000	67,500	64,150	60,300	56,650
1987	"	8050	87,350	79,050	75,550	72,200	68,350	64,700
1988	"	8200	95,550	87,250	83,750	80,400	76,550	72,900
1989	"	8300	103,850	95,550	92,050	88,700	84,850	81,200
1990	"	8350	112,200	103,900	100,400	97,050	93,200	89,650
1991	"	"	120,550	112,250	108,750	105,400	101,550	97,900
1995	"	"	153,950	145,650	142,150	138,800	134,950	131,300
2000	"	"	195,700	187,400	183,900	180,550	176,700	173,050
2005	"	"	237,450	229,150	225,650	222,300	218,450	214,800
2010	"	"	279,200	270,900	267,400	264,050	260,200	256,550
2020	"	"	362,700	354,400	350,900	347,550	343,700	340,050
2030	"	"	446,200	437,900	434,400	431,050	427,200	423,550
2040	"	"	529,700	521,400	517,900	514,550	510,700	507,050

TABLE 7-II. ACCUMULATED OPERATING HOURS TWO AND THREE EQUAL LIFTS

Other aspects of operation applicable to all concepts such as frequency of starts and record-keeping are presented in Chapter 6F.3. on single-lift operation.

4. Maintenance

It is assumed that corresponding units at each plant will be maintained simultaneously to minimize plant outage time. Two repair crews per shift are needed for this type of operation.

Predicted repair times for the two-lift concept are: shaft packing repair every 13,900 pump hours and pump overhaul every 31,200 hours. Repair time is assumed to be 50 hours for packing and 250 hours for overhaul.

Shaft packing is used on both ends of the pump. Overhaul involves the probability of replacement of four wear rings and two interstage seal rings. Welding of guide vanes and repair or replacement of two suction impellers may also take place at this time. The prototype design shows the rotating section of the wear rings to be an integral part of the imepllers and four throttling surfaces are involved. More comments on shaft packing maintenance may be found in Chapter 6F.4. and Table 6-V.

Motor maintenance should be quite similar to that outlined in Chapter 6F.4. for the single-lift plant, except that if there is any real correlation between number of starts and motor life, winding repairs might be more frequent. Frequency of starts per unit in the two-lift will probably exceed that of the single-lift because of interdependent operation.

Maintenance considerations for the two-lift concept pump discharge valves are the same as discussed in Chapter 6F.4. for the single-lift concept. Repair of the pump inlet butterfly valves for the second plant of the two-lift concept may be distinguished from repair of the spherical valve in that replacement or inspection of its seat requires de-watering of adjacent piping. Otherwise, all associated equipment requires much the same care.

Other comments on maintenance applicable to all lift concepts may be found in Chapter 6F.4. on single-lift maintenance. Analyses of pump repair times are given in Volume II, Chapter 7.

5. Reliability

If the plants of the two-lift concept could be made independently operable, the concept would be more reliable than the single lift due to inherent thrust balancing, but restrictions imposed by one upon the other make the concept less reliable than single lift though more reliable than three lift.

It was pointed out in Chapter 6F. 5. that since both ability and availability to operate are necessary, the product of the two must be used in determining overall concept effectiveness or reliability. Similarly, since both plants of the two-lift concept must be able to operate, their overall ability to operate is the product of their individual abilities.

The effectiveness index which measures chances for successful accomplishment of water deliveries is 0.950 for the two-lift, while the single lift is 0.979 and the three lift is 0.902. Derivation and significance of these factors are further explained in Chapter 7 of Volume II and in Chapter 6F. 5. of this volume. It must be reiterated here that these figures are not representative of absolute concept effectiveness but are valid indications for rating purposes as far as can be determined at present.

CHAPTER 8

THE THREE-LIFT SYSTEM

A. PROTOTYPE PUMPS

1. Design Conditions

In order to use single-stage pumps, at least three separate pump stations are necessary to make the total lift. The total dynamic lift of 1961 feet requires three pumps in series producing 650 feet of head each. Each plant would have nine such pumps with a capacity of 555.6 cfs to meet an assumed total flow of 5,000 cfs.

The Byron Jackson Co. of Los Angeles was selected to perform prototype design and model testing of a single-stage pump.

Specified conditions for the pump and pumping plant are given below:

a. Pump Specifications

(1)	Total Dynamic Head, H	650 feet.
(2)	Flow Rate, Q	555.6 cfs
(3)	Rotating Speed, N	514 RPM
(4)	Stages	1
(5)	Mounting	Vertical, Axial Inlet
(6)	Submergence	To be specified - (see A. 4. of this chapter).
(7)	Permissible Discharge Pressure Rise	20% of normal
(8)	Service	Water pumping, continuous duty.

b. Plant Conditions

(1) Geographical Location Zone V - California Plane Coordinate System.

(2) Plant Elevation

Plant 1 - Approx. 1245 ft. ground elevation

Plant 2 - Approx. 1880 ft. elevation

Plant 3 - Approx. 2520 ft. elevation

(3) Pumping Pond Elevation - Plant 1

Normal	1239 ft.
Minimum	1229 ft.
Maximum	1240.5 ft.

(4) Terminal Canal Elevation - Plant 3

Normal	3167.6 ft.
Minimum	3162 ft.
Maximum	3173 ft.

(5) Water Inlet Passages

8-1/2 ft. diameter x 30 ft. in length

(6) Discharge Valves - Spherical Valve

(7) Motor Drive - Electrical - 13.8 KW synchronous, 60 cycle

(8) Water Quality - To be determined later.

c. Transient Conditions

(1) Start-up - Full voltage starting is planned with the pump casing filled with water.

(2) Normal Shutdown - Controlled valve closure.

2. Pump Design and Description

The cross-section of the prototype pump is shown in FIG. 8-1, with concrete embedment indicated. An outline drawing is presented in FIG. 8-2 and the suction piece detail is shown in FIG. 8-3. Excerpts from the Byron Jackson description are given below. The complete B-J write-up is given in Vol. II, Chapter 12.C.

"Components

Drawing 1F-6554 {FIG. 7.1} "TEHACHAPI PROTOTYPE DESIGN", shows a detailed section through the pump with critical dimensions and running clearances.

(1) Impeller

Impeller Diameter 88-15/16

Eye Diameter 50-3/4"

$$\frac{\text{Eye Dia.}}{\text{0. Dia.}} = \frac{50.75}{88.94} = .571$$

The overhung impeller is balanced with compensation for the suction pressure. The balanced area on the back shroud is piped back to the suction side of the pump.

The impeller is taper mounted, and driven with a double key. Critical stresses in the impeller will be calculated after the final hydraulic design has been completed and verified by tests.

(2) Case

The volute case is fitted with integral diffuser vanes. In addition to forming the hydraulic case passages, these vanes serve the following purposes:

(a) Acts as stiffening ribs against deflection in the case, permitting lighter case walls for a given pressure.

(b) Give a more uniform pressure distrubition around the impeller, thereby, reducing radial thrust.

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

(c) Reduce pressure fluctuations over the specified operating range.

(3) Stuffing Box

The stuffing box is an integral part of the combination bearing bracket and back cover. It is fitted with a spiral grooved breakdown bushing, working against a rotating, grooved shaft sleeve. With the proper combination of running clearance, groove shapes and relative pitch, leakage will be controlled to the desired level to produce adequate flushing.

(4) Bearing Bracket

The bracket is designed with a minimum distance between the centerline of the bearing and the centerline of the impeller.

The self-aligning spherical sleeve bearing is pressure lubricated from a central oiling system.

(5) Suction Piece

Drawing 406031, {FIG. 8-3} "SUCTION PIECE", shows a section through the taper elbow, connecting the 8-1/2 ft. diameter inlet passage to the pump suction flange. The purpose of combining the required tapered section and the 90° bend is to create an accelerating elbow which will give uniform velocity distribution in the impeller eye. This prevents premature cavitation, caused by a local pressure drop.

Drawing 722336-1 {FIG. 8-4} shows typical velocity distribution at the small end of an accelerating elbow as established by tests. There was no backflow between 85% and 130% of design capacity, and consequently, no pre-rotation.

"Measures Taken to Insure Highest Efficiency and Reliable Operation

(1) Efficiency

(a) The overhung design permits an optimum hydraulic design for the suction elbow, since the need for a bearing with a restricting bearing span on the suction has been eliminated.

(b) No hydraulic disturbances from a rotating shaft or sleeve exists in the impeller eye.

(c) With elimination of the shaft through the impeller eye, the front wear ring diameter can be held to a minimum for a given impeller eye area. This in turn minimizes leakage losses at the front wear ring.

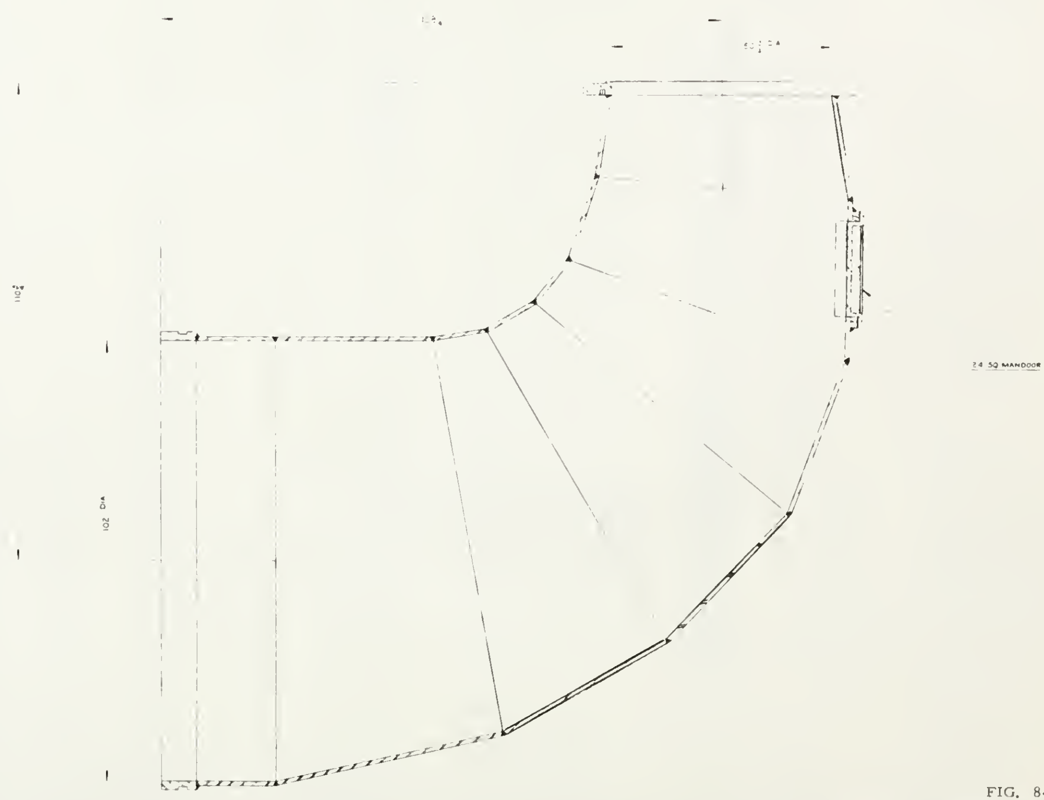


FIG. 8-3

D M J & M CONTRACT No 6371-C

BYRON JACKSON ORDER No 722336 L

SUCTION PIECE
TEHACHAPI PROTOTYPE PUMP

BYRON JACKSON DIVISION

406031

(From Byron Jackson Design Report)

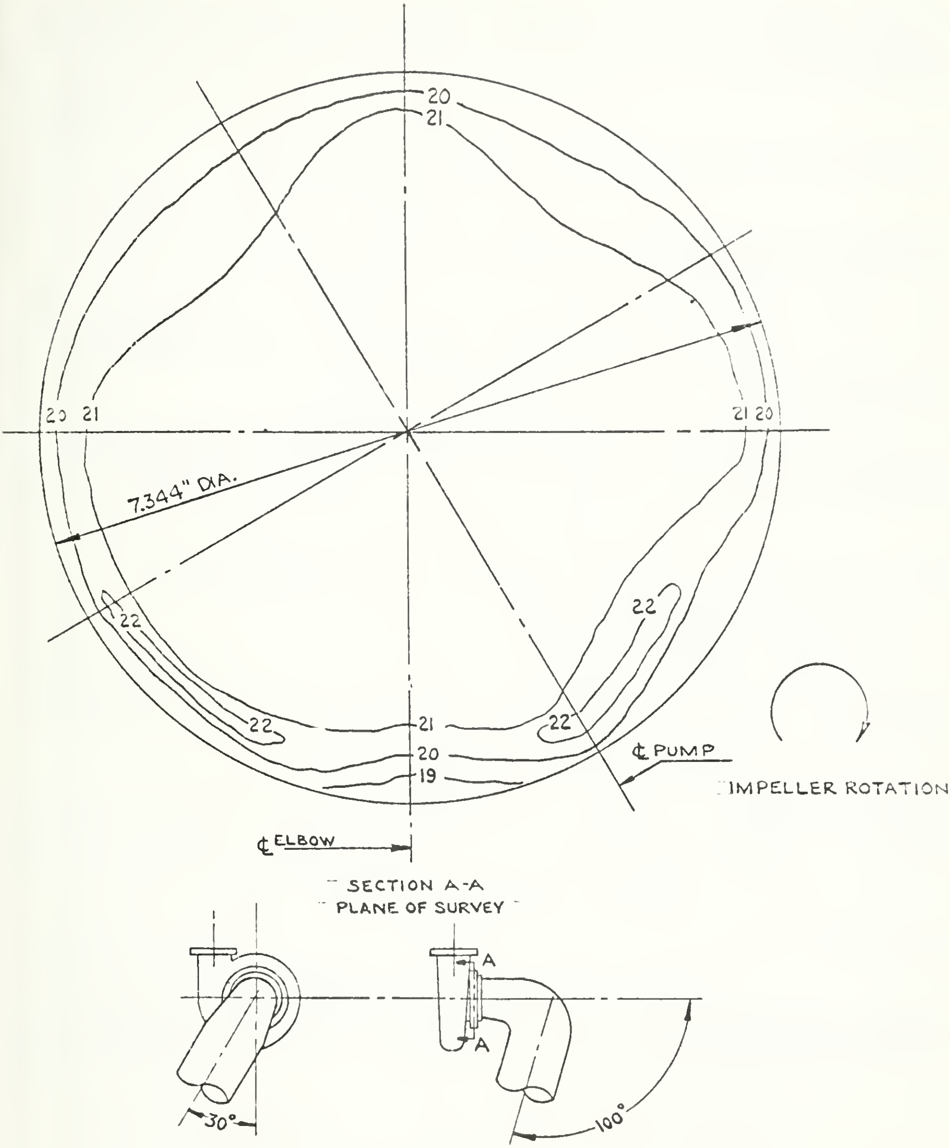


FIG. 722336-1

VELOCITY DISTRIBUTION IN ACCELERATING ELBOW
(FT. PER SEC.) FLOW = 6.41 CFS

FIG. 8-4

(d) The closeness of the overhung impeller to the bearing, combined with a heavy shaft, results in rigidity and small shaft deflections. This permits the use of close running clearances with minimum leakage losses.

(e) The use of properly designed grooving on wear ring surfaces and throttle bushing further reduces leakage losses.

(f) Simplicity of design results in easily accessible hydraulic passages throughout the pump. These can be given a uniformly good finish, resulting in low hydraulic friction losses.

(g) Simplicity of design also permits the use of narrow clearances between impeller shrouds and pump covers. This results in a substantial reduction in disc friction losses.

(h) The elimination of internal self-lubricated bearings and the use of just one external oil lubricated bearing in the pump, holds the mechanical friction losses to a minimum.

(i) The overhung design gives full freedom to use the latest developments in the art of hydraulic impeller and case design.

(2) <u>Reliable Operation</u>

(a) The uniform velocity distribution pointed out in Paragraph 4. 1. 1 and 4. 1. 2 protests against unexpected localized cavitation, thereby insuring long impeller life.

(b) The rigid shaft and bearing design prevents any metal to metal contact in areas with close running clearances under all operating conditions. This will maintain high pump efficiency under sustained operation.

The pump may be operated "dry" without damage for extended periods.

(c) There are only two (2) alignment fits from the bearing bracket to the front pump cover. With each one located in relation to the impeller with feeler gauges and doweled in place, virtually perfect alignment is assured with minimum mechanical vibration or wear.

(d) Seal water connections to wear rings and breakdown bushing permits flush-out of abrasive sediment prior to start-up.

(e) With the low wear rate at the impeller balance ring and front skirt, we propose to eliminate impeller wear rings. At such time when the leakage loss has become excessive, it will cost less in material and labor to replace a case wear ring and take a skim cut on the impeller wear surfaces, than to replace a set of wear rings.

(f) With elimination of a sealing contact in the stuffing box, wear and maintenance is reduced.

(g) The external self-aligning, oil lubricated bearing results in minimum bearing friction and long life. Temperature and pressure detectors connected to shut-down relays, protects the pump against damage from failure of auxillary equipment.

(h) By disconnecting the suction piece and front pump cover, the design permits removal of the impeller, the case wear rings and the breakdown bushing through the bottom of the pump without disturbing the bearing bracket or the driver. The split bearing assembly can be removed from the top without disturbing the driver or the pump proper.

"<u>Operation Under Transient Conditions</u>

(1) Until the Engineer furnishes a preliminary analysis of the transient conditions, as stated in Section 3.02 b of the specification, this discussion will be limited to a brief description of start-up and shut-down procedures.

(2) <u>Start-up Procedure</u>

(a) The unit will be started under full voltage (13.8KV-60) with the pump case full of water and the discharge valve closed.

(b) The seal water connections to the wear rings and the stuffing box breakdown bushing, and the gravel and blow-off valve will be opened to flush sediment out of the close running clearances.

(c) The oil pump, providing lubrication for the prototype pump bearing, will be started. When the prescribed bearing pressure has been reached, a pressure switch will close and permit starting of the prototype motor.

(d) The prototype, synchronous motor can now be started. This will automatically close seal water connections and the blow-off valve.

When synchronous speed has been reached, controlled opening of the discharge valve will automatically start. See attached FIG. 722-366-2 {FIG. 8-5} for starting torque requirements.

(e) Temperature sensing devices in the prototype pump bearing as well as in the discharge line between the pump and the discharge valve will automatically shut the unit down if abnormal temperatures occur.

(3) Shut-Down Procedure

(a) The first step will be controlled closure of the discharge valve.

(b) The full closure of the discharge valve will automatically cut the power to the synchronous motor and shut down the unit.

(c) A time delay relay will shut down the oil pump after the unit has come to a stop.

(4) A complete analysis of pump behavior under transient conditions will be given after the three (3) quadrant tests have been performed.

3. Pump Performance

The Byron Jackson Company bases their performance prediction on extensive testing of their Grand Coulee model pump which is very similar to the Tehachapi pump. Their discussion of efficiency and "step-up" is given in Volume II, Chapter 12-C. Abstracts from the B-J report and other comments follow:

a. Specific Speed

The specific speed as determined by the specified operating condition is N_s = 1990 (note that B-J rounds off to 2000 in their report). In a separate study of Specified Speed Vs. Efficiency, B-J has stated that high pump efficiency in general will be obtained with $1800 < N_s < 3500$ and that considering other design features of the Tehachapi application, the range of 1800 to 2400 is best. Therefore, the selected N_s of 2000 is proper.

b. Efficiency Prediction

B-J states the model efficiency to be 91.0% $\pm$ .3 and this figure is presumably based on Grand Coulee model tests. The step-up formula employed for full head model testing is:

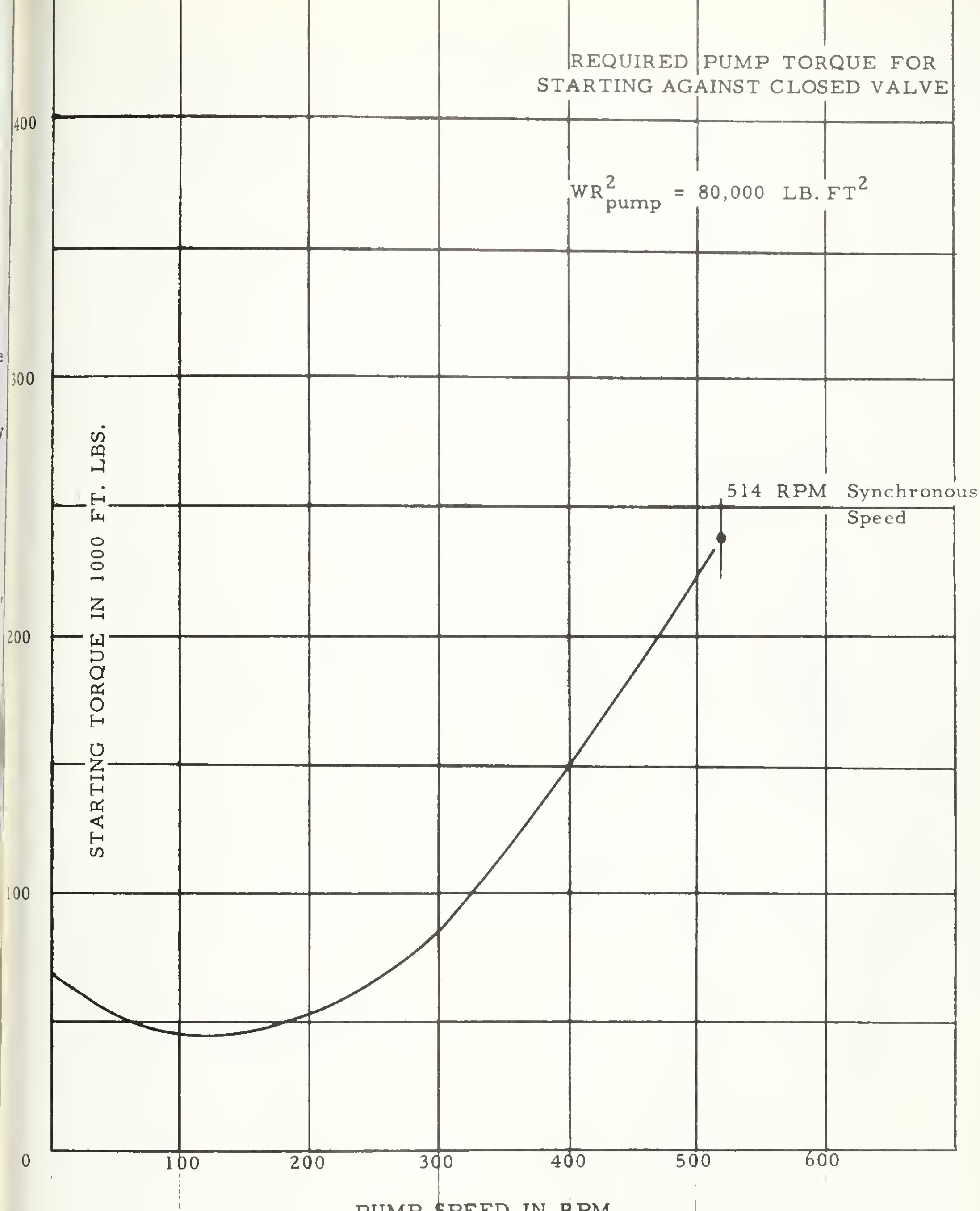

REQUIRED PUMP TORQUE FOR STARTING AGAINST CLOSED VALVE
WR^2_{pump} = 80,000 LB. FT^2
514 RPM Synchronous Speed
STARTING TORQUE IN 1000 FT. LBS.
400
300
200
100
0
100
200
300
400
500
600
PUMP SPEED IN RPM

FIG 8-5

$$1 - \frac{\eta_{proto.}}{\eta_{model}} = \left[\frac{D_m}{D_p}\right]^n \qquad (8\text{-}1)$$

where n is taken as 0.165 for B-J model tests. B-J points out that their exponent of .165 is more conservative than values of n = .20 and .25 found in the literature. The resultant prototype efficiency prediction is:

$$\eta = 93.2\% \pm .3$$

(See Volume II, Chapter 12.C for a lengthy discussion by B-J and Volume II, Chapter 8 for an analysis of "Step-up").

c. Performance Curve

Predicted performance for the pump is presented in FIG. 8-6.

4. Cavitation Characteristics and Submergence Requirements

Byron Jackson has suggested a design range of Suction Specific Speed of 8000 to 9000. Taking S = 8000, then NPSH = 102. This would correspond to a submergence of 70.7 ft. at the first pumping plant.

Submergence conditions for S = 7000 and S = 8000 are given below:

	S	NPSH	Plant No.	Sub-mergence	Normal Intake Water Level	Elevation of Pump
Per Byron Jackson	8000	102	1	70.7	1239	1168.3
			2	71.5	1881	1809.5
			3	72.2	2524	2951.8
DMJM Tentative Recommen-dation	7000	121	1	89.7	1239	1149.3
			2	90.5	1881	1790.5
			3	91.2	2524	2432.8

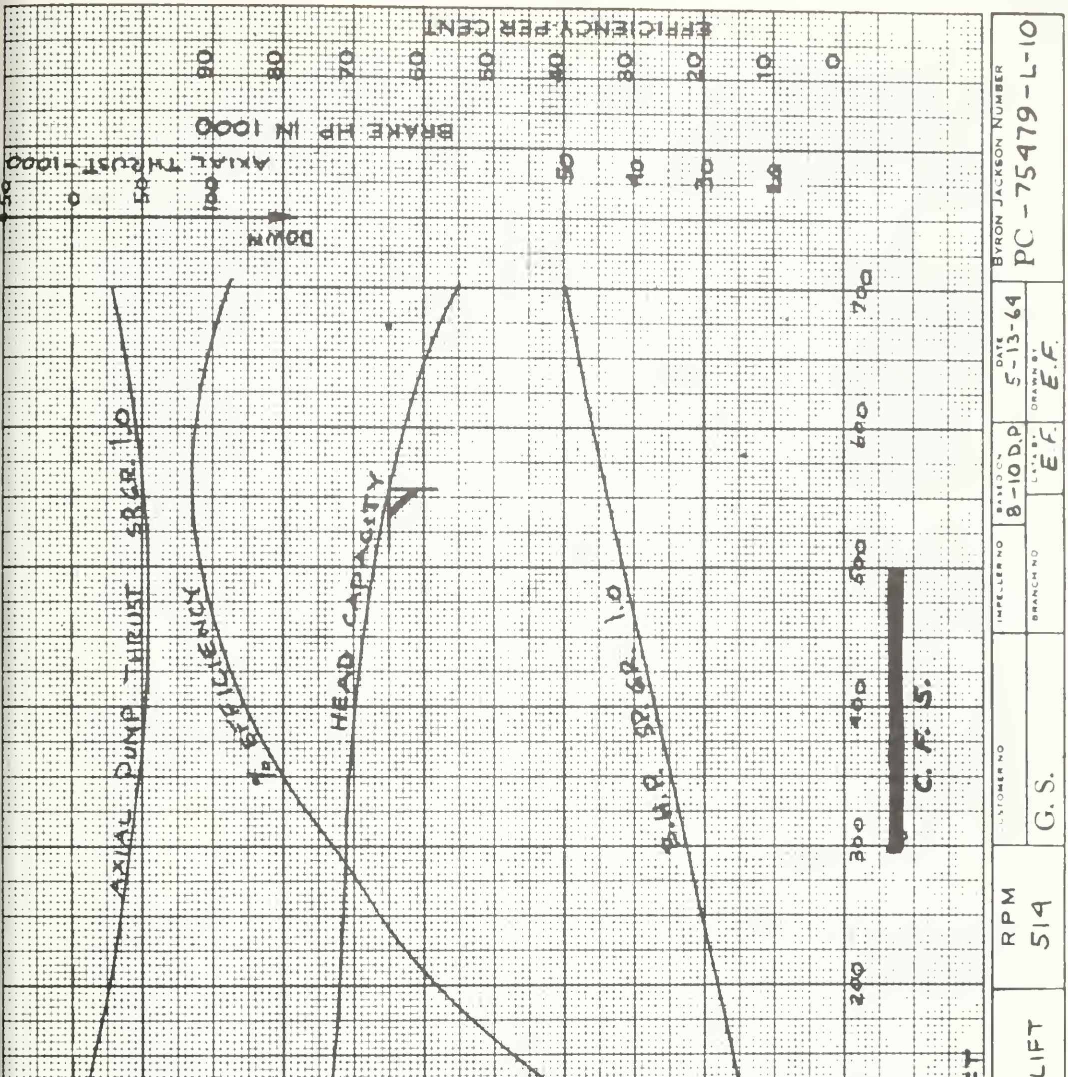

EFFICIENCY PER CENT
BRAKE HP IN 1000
AXIAL THRUST-1000
DOWN
AXIAL PUMP THRUST SP.GR. 1.0
EFFICIENCY
HEAD CAPACITY
B.H.P. SP.GR. 1.0
C.F.S.
9 UNITS PER LIFT
TOTAL HEAD IN FEET
1 STG. VERT S.S- 3 LIFT
RPM
514
G.S.
8-10D.P
DATE
5-13-64
E.F.
E.F.
BYRON JACKSON NUMBER
PC-75479-L-10

Fig. 8-6

If the water level in the forebay of the first plant dropped to the low water limit of 1229 ft., the submergence would be reduced 10 ft. which would increase the value of S for the pumps by about 500. For this reason, a normal value of S = 7000 would be more desired than a normal value of 8000.

It must be noted here that the Grand Coulee pump has been operating at higher flow and lower head than the b.e.p.[1] With operation on the right of the H-Q curve, the NPSH requirement is higher than at b.e.p. The submergence has also been greater than considered normal in the design stage and the resultant operating S is around 4500 to 6000. Yet, the impellers have been suffering cavitation damage. It is difficult to evaluate off-design-point cavitation performance because blade entrance angles are incorrect, and under these circumstances local cavitation is to be expected, even with relatively high values of submergence. It is possible to have conditions of high flow rate (greater than b.e.p.) at Tehachapi and the same problem could occur. In any event, cavitation erosion with high values of S is possible, even though pump head and efficiency are not noticeably effected. Thus, S = 7000 is recommended by DMJM until model tests can show a justifiably higher value.

5. Pump Materials

The B-J tentative selection of materials is cast carbon steel for the case and diffuser and bronze for the impeller. Specific alloys and auxillary parts have not been specified per ASME Unfired Pressure Vessel Code in the stress calculations.

6. Pump Stresses

The complete stress analysis submitted by Byron Jackson is given in Volume II, Chapter 12C. Results are given below and are considered preliminary, subject to calculations following model tests:

a. Drive key stress = 11,150 psi.

b. Volute wall "hoop" stress = 14,850 psi, ASME Unfired Pressure Vessel Code allows 17,500 psi working stress for SA-216WCB (Table WCS 23).

c. Diffuser vanes are subject to a total force of 7,374,000 lb. requiring a total vane cross-sectional area of 635 in^2 for 17,500 psi stress.

d. With a shaft diameter of 18" the stress to carry the torque is 4720 psi.

---

[1] best efficiency point

e. With closed discharge, an estimate of the radial load is 24,650 lbs. The corresponding shaft deflection would be .0002 in.

f. The critical speed is calculated to be 2980 RPM, considerably above the 514 RPM operating speed and conceivable over-speed.

g. The radial bearing load at closed valve is 64.3 psi.

7. Cost and Weight

The three lift concept requires nine (9) pumps in each station, resulting in a total of 27 pumps. With the pumps purchased in lots of 9, the cost is:

Unit Cost	=	$ 136,500
Total cost of 27 pumps	=	$3,685,000

The unit weight is 130,000 Lb. (65 tons).

B. MODEL TESTING PROGRAM

1. Model Testing Firm

The testing of the model pump for the three-lift system will be conducted by the manufacturer, Byron Jackson Pumps, Incorporated, a subsidiary of the Borg-Warner Corporation. The location of the main plant and testing facilities is 2301 East Vernon Avenue, Vernon, California.

2. Description of Testing Laboratory

The Byron Jackson Hydraulic Test Facility occupies a section 153 x 78 feet of the pump manufacturing facility, located in the Vernon Plant. This facility is highly specialized in performing tests appropriate to large centrifugal pumps. Approximately 25 percent of the laboratory area is built directly over the main test tank, a below surface reinforced concrete structure. The tank is 30 feet wide, 102 feet long, a depth range of 10 to 30 feet, and a capacity of approximately 400,000 gallons. Its floor covering is divided into twenty 5-1/2 x 10 foot openings, permitting concentrated loading. Heavy equipment is moved to different positions in the laboratory by means of two overhead cranes, 15-ton and 40-ton capacities. The laboratory is served electrically with two transformer banks, 1500 KVA, 2300/580 V.A.C., 50 or 60 CPS with switch gear for each voltage. FIG. 8-7, Byron Jackson Drawing Number 1C-2430, Revision B, shows the test stand configuration for the model pump tests. The specific test equipment and instrumentation to be employed is described in Volume II, Chapter 1.

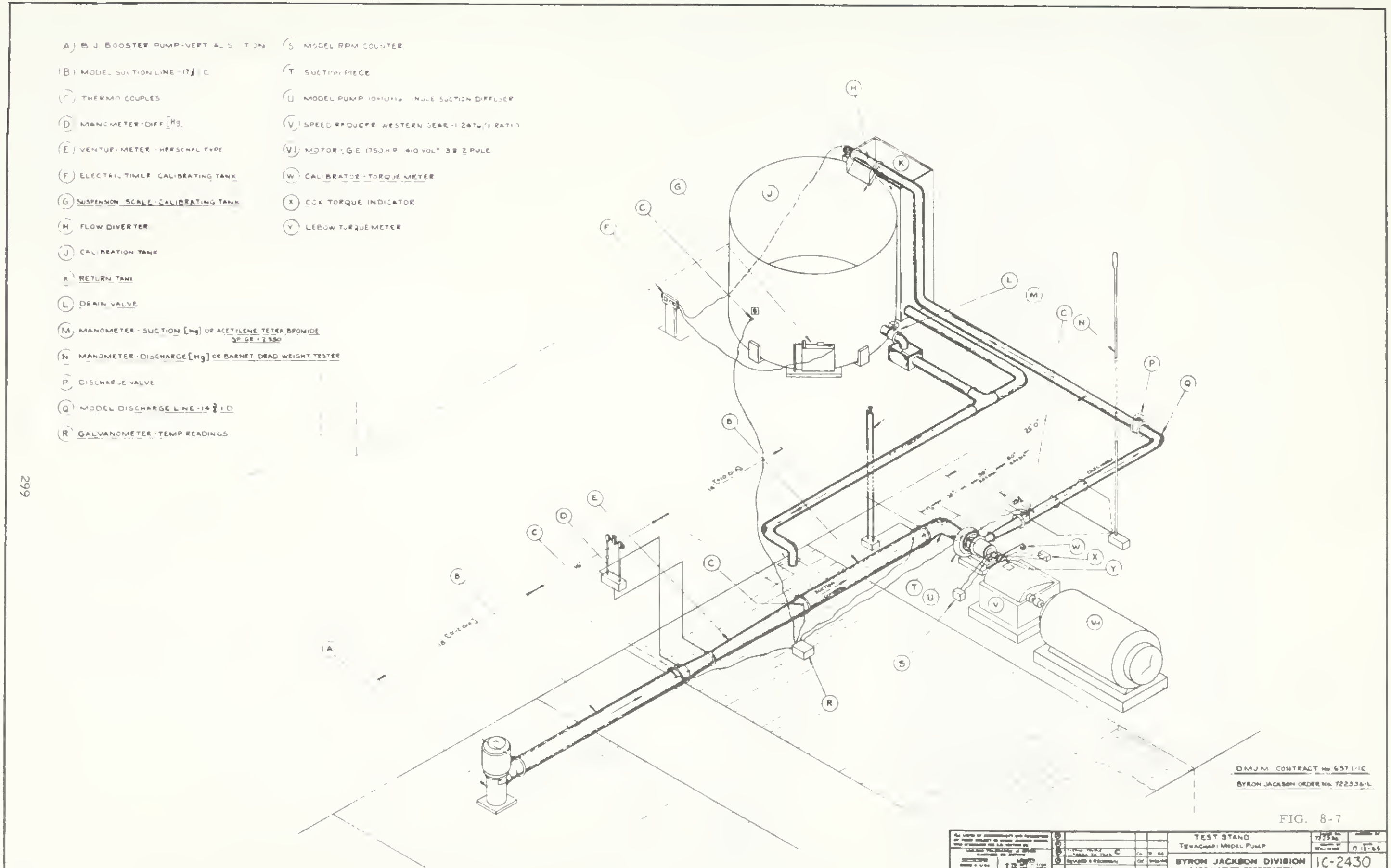
3. Model Pump

A cross sectional view of the Byron Jackson model is shown in FIG. 8-8. Although the inlet section is not shown, it will be a scale model of the prototype reducing elbow shown in FIG. 8-3.

During preliminary testing, Byron Jackson plans to test two impeller designs and two diffuser ring designs in all four possible combinations.

4. Model Pump Test Procedure

Byron Jackson has submitted detailed descriptions of tests and test procedures. These are presented in Volume II, Chapter 1.



A) B.J. BOOSTER PUMP-VERT.
B) MODEL SUCTION LINE
C) THERMO COUPLES
D) MANOMETER-DIFF [Hg]
E) VENTURI METER - HERSCHEL TYPE
F) ELECTRIC TIMER - CALIBRATING TANK
G) SUSPENSION SCALE - CALIBRATING TANK
H) FLOW DIVERTER
J) CALIBRATION TANK
K) RETURN TANK
L) DRAIN VALVE
M) MANOMETER - SUCTION [Hg] OR ACETYLENE TETRA BROMIDE SP. GR. = 2.950
N) MANOMETER - DISCHARGE [Hg] OR BARNET DEAD WEIGHT TESTER
P) DISCHARGE VALVE
Q) MODEL DISCHARGE LINE
R) GALVANOMETER - TEMP READINGS
S) MODEL RPM COUNTER
T) SUCTION PIECE
U) MODEL PUMP
V) SPEED REDUCER WESTERN GEAR
V1) MOTOR - G.E.
W) CALIBRATOR - TORQUE METER
X) COX TORQUE INDICATOR
Y) LEBOW TORQUE METER
DMJM CONTRACT No. 637-1-IC
BYRON JACKSON ORDER No. 722336-L
TEST STAND
TEHACHAPI MODEL PUMP
BYRON JACKSON DIVISION
BORG-WARNER CORPORATION
IC-2430

FIG. 8-7

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

5. Predicted Model Pump Efficiency

Byron Jackson predicts that the model pump efficiency will be $91.0 \pm 0.3\%$. A complete discussion of this point can be found in Chapter 12, Volume II.

6. Test Schedule

The schedule of test event is being followed by DMJM through a Program Evaluation and Review Technique (PERT) Control System. Information is received from Byron Jackson on the progress of each event on a bi-monthly basis. This information is processed by DMJM, injected in the PERT system for the entire program and forwarded to the Department of Water Resources. Abstract sections of the PERT Chart are sent to Byron Jackson, advising them of the current status. A discussion of the PERT system is presented in Chapter 2, Volume IV.

7. Test Results

Slow speed, preliminary tests have been completed just prior to preparation of this report and Byron Jackson has submitted a model efficiency of 91.2 for use in the comparative analysis. Stepped up prototype efficiency and discussion are given in Chapter 9.

C. MOTOR

Except as noted below, the motors for the three-lift system, although of much lower horsepower rating, are essentially the same as the motors for the single-lift and two-lift systems as set forth in Chapters 6C and 7C. It is also assumed that the three-lift motors will be self-starting and can be built to reliably meet the requirements set forth below for the Motor Prototype Design. The general arrangement for a typical motor is indicated on Drawing E-1, {FIG. 6-15} .

1. Motor Prototype Design

The nine main pump motors at each of the three plant are 46,000 horsepower, 514 RPM, 13,800 volt, 1.0 power factor vertical synchronous type and designed for 40% overspeed.

2. Motor Starting

Motors will be started at full voltage by the same method as used for the single-lift and two-equal lift pump motors. The required starting duty however for the motors of the three-lift system is appreciably less than for the corresponding higher speed and greater horsepower motors of the single-lift and two-lift systems. It is estimated that the starting inrush at rated voltage for the 46,000 horsepower motors will be approximately 130,000 KVA which compares with values of 197,000 KVA for the two-lift, 70,000 horsepower motors and 243,000 KVA for the single-lift 80,000 horsepower motors.

3. Power Requirements and Efficiency

An efficiency value of 98.4% has been assigned to the 46,000 horsepower motors which is the same value used for the larger motors required for the two-lift and single-lift systems. This will result in a power requirement for each motor of approximately 35, 100 KW with the motor delivering rated horspower or approximately 315,900 KW at each of the three stations, exclusive of power system losses and the supporting station load requirements. For the purpose of analyzing motor performance, it is assumed that the nine main pump motors at each of the two stations will be supplied by three main power transformers of 110,000 KVA capacity with three motors connected to each transformer.

Based on an estimated starting time of 30 seconds and 51% of rated horsepower being required at rated speed while operating against a closed head, the average rate of power increase during the motor starting period will be approximately 600 KW per second which at the end of the 30 second accelerating period will equal 18,000 KW. This will be followed by the gradual opening of the discharge valves to bring the pump and motor up to a steady state rated load condition.

D. VALVES

1. Design and Operation

The rate of flow through each pump and discharge valve is 556 cubic feet per second for the three lift as on the two lift concept. A 63-inch inside diameter valve is appropriate for either multilift concept. The discharge pressure for the three lift concept with equal lifts is approximately 370 psi, compared with 550 psi on the two lift concept, allowing in each case 25% for overpressure due to pressure surge and pump shut-in pressure.

Operating head for the three lift concept is approximately the upper limit for normal application of butterfly valves in the discharge function. More suitable, perhaps, are spherical, needle, or cone plug valves as described in Chapter 6.D. and in the Technical Volume.

Butterfly valves, approximately 8.5 foot diameter, would be appropriate for the inlet valve to pumps on upper lifts without forebays. The design pressure for the inlet valve would be 50 psi nominal to accommodate 90 or more foot submergence or inlet head. These valves are discussed in Section 7.D. and in the Technical Volume together with sluice valves.

The pump discharge valve on the three lift concept should be capable of opening or closing with 370 psi differential across the valve at the start of the opening cycle and the end of the closing cycle. The closing cycle of the valve should be adjustable to close the first 80% of the port area in from 10 to 20 seconds and the last 20% of its port opening in 20 to 40 seconds. Seating operations involving a small percentage of full flow may be exclusive of closing time requirements.

2. Size, Weight and Costs

The spherical valve operator occupies a position adjacent to the valve, where the trunion extends through the valve body. The hydraulic cylinder operator extends as much as three feet past the valve body, resulting in an overall width of 11 to 12 feet and a length of 7 to 9 feet for a 63-inch spherical valve. The overall height from the floor to the top of the valve may be 12 or more feet, depending on the point of attachment for the other end of the cylinder. The shutoff valve would weigh about 135,000 pounds. The inlet butterfly valve may be from 2 feet to 3 or more feet long and approximately 13 feet wide including operator. Manufacturer recommendations suggest the use of metal seats for a valve this large. The butterfly valve would weigh approximately 40,000 pounds.

The cost of a spherical valve 63-inch inside diameter and 370 psi design is approximately $170,000 with operator and hydraulic supply unit. The 8-foot - 6-inch butterfly valve with operator costs approximately $60,000.

E. PLANT ARRANGEMENT

A cross-section of a pumping plant corresponding to Plant 2 or 3, is shown in FIG. 8-9. Plant 1 would be the same, except the inlet butterfly valve would not be necessary.

In this installation, the pump volute is embedded in concrete. This is unlike the other two pump types, which can be removed from their foundations by disconnecting the inlet and discharge piping.

The Byron Jackson Company has suggested a removable inlet station permitting disassembly of the pump through the inlet of the pump. FIG. 8- 9 shows this possibility with the entire inlet section removable. It might be more desirable to split the volute into two sections with the lower section embedded in concrete. It might also be more desirable simply to embed the inlet section in permanent assembly with the pump case, and do all overhaul work from the top side as is conventional with single-stage pumps in this type of service. A very minor modification in the B-J pump mechanical design would permit this, and a decision on this matter can be made when more detailed consideration is given to the plant layout.

The single-stage pump is designed with hydraulic balancing by using nearly equal-diameter wear rings on both sides of the impeller and providing an inlet pressure connection to the shaft side of the impeller. As with the other lift types, this pump and motor are vertical with an intermediate shaft. The thrust bearing is built into the motor and must support the weight of all the rotating parts.

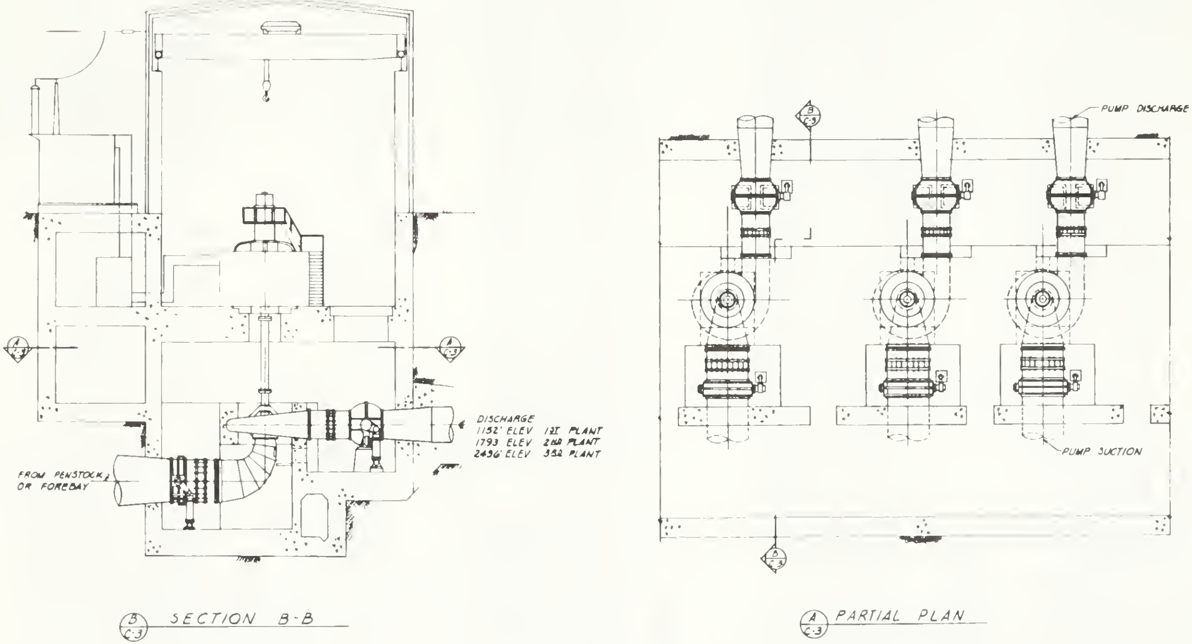


FIG. 8-9

DMJM DANIEL MANN JOHNSON & MENDENHALL
3325 WILSHIRE BLVD. LOS ANGELES 5, CALIFORNIA DUNKIRK 1 3663
PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS

THE RESOURCES AGENCY OF CALIFORNIA
Department of Water Resources

TEHACHAPI PUMPING PLANT
THREE - LIFT SYSTEM

TYPICAL UPPER PUMP STATION
SECTION & PARTIAL PLAN

637-1-1
C-3
MARCH 5, 1960

F. OPERATION AND MAINTENANCE

1. System Elements

Elements identifiable with the three-lift concept in regard to operation and maintenance are its three-plants-in-series operational mode and its simplicity of pump design which affords shorter repair times than multistage lift concepts. Motors and valves are of a more conventional nature than those employed in the other lift concepts.

2. Precedent

Single stage pumps are commonly used in high capacity plants in the United States. Eleven (11) U. S. pumping plants, all with single stage pumps, were visited for this study and six (6) of the European plants surveyed used single stage pumps. (See Chapter 2 of Volume II and Volume III). Of the seventeen single stage pumps evaluated in the study, twelve were of the vertical variety, thirteen utilized single suction, two operated with a head above 450 feet, and, in one case, a multilift-without-forebay concept was in use.

3. Operation

Operation of a three-lift plant without forebays requires that each of two plants shut down a unit when the third plant stops a unit. As with the two-lift concept, this mode of operation leads to a lower reliability than if the plants could operate independently, in which case the three-lift concept would be most reliable. The problems of operational control and decision-making as mentioned in Chapter 7.F.3 for the two-lift concept are further complicated by the added lift.

Discussions of Tehachapi operations in general, including starting frequency and record-keeping, may be found in Chapter 6.F.3. Table 7-I in Chapter 7.F.3 gives projections of water demand buildup and accumulated unit operating hours to year 2040.

4. Maintenance

It is assumed that identical scheduled maintenance actions will be performed at each plant in the three-lift concept simultaneously. Because there are only two rings, one seal, and one impeller per unit, overhaul time is normally shorter for single stage pumps than multistage. However, in all probability, rings will wear more quickly in the three-lift concept than in the others because the head per stage is about 650 feet as opposed to 490 in the other two concepts.

Preliminary estimates of repairs indicate the shaft packing may be replaced every 12,900 operating hours in a 50-hour repair period and that overhaul might be needed every 29,000 hours and take about 230 hours of repair time. Only one packing need be maintained for this concept instead of two as with the others. Two rings and one impeller must be maintained at overhaul but the rings will probably not be as simple as those found in most U.S. pumps because of the higher operating head. Analytical back-up for maintenance time predictions is given in Volume II, Chapter 7.

Motors will be of smaller size and possibly easier to re-wind but mean times between maintenance actions might be adversely affected by an increased number of starts associated with series plant operation.

Maintenance considerations for the pump discharge valves in the three-lift concept are the same as those discussed in Chapter 6.F.4 for the single-lift concept. Maintenance considerations for the three-lift concept pump inlet valves are the same as discussed in Chapter 7.F.4 for the two-lift concept. Other comments on maintenance applicable to all lift concepts may be found in Chapter 6.F.4 on the single-lift concept.

5. Reliability

In establishing the comparative reliability of the lift concepts for successful delivery of water, the three-lift scored 0.902, as compared to .979 for the single-lift and 0.950 for the two-lift. Even when an assumed overhaul time for multistage pumps is double that for single stage pumps, the same general results are obtained. Additional comments on reliability of lift concepts are given in Chapter 6.F. and 7.F. of this volume, and Chapter 7 of Volume II contains detailed examples and explanations of the reliability analysis.

CHAPTER 9

COMPARATIVE ANALYSIS

A. DATA:

In Table 9-I, an attempt has been made to present all data on prototype pumps and interface items that is pertinent to the selection of the lift concept. Unfortunately, there is such a vast amount of data that it cannot readily be placed into one presentation and many of the details are to be found in the specialized sections of the report. Further, the choice of vendors cannot logically be made at this time, so some of the data must be presented in "ranges". Where different manufacturers give wide variations in the technical performance of components and of the costs, it is not easy to sum component information which will give the total plant performance and cost.

The experimental work with the model is a very important part of the Tehachapi study. Unfortunately, the model program has many months of work remaining and only preliminary results are available at this time. The preliminary model test results are given in Table 9-II.

B. DISCUSSION:

The principal considerations for the pump and its interface item selections that fall within the DMJM scope of work are:

Reliability
Pumping Efficiency
Initial Cost
Submergence Requirements

The listing order is thought to reflect the order of importance.

The overall evaluation must include these factors, plus costs and reliability of the elements of the lift The efficiency can be related to total lifetime operating costs in order to obtain the comparative total economy.

Another important consideration is motor starting, but for this study it is assumed to have equal effect on all three lift concepts.

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

1. Comparative Reliability

An analysis of the reliability of the three distinct lift concepts was conducted to provide comparative data for selection of the optimum pump system. In deriving comparative reliability, emphasis was placed on the pump and pump components. Further studies of the selected lift concept will be conducted as this program progresses, to provide absolute reliability. A final value of absolute reliability is required so that operational and maintenance procedures may be developed. Comparative reliability, however, will provide a portion of the fundamentals for selection of the proper lift concept.

On the basis of the comparative reliability study performed in this program, the single lift concept must be rated first, the two-lift concept is rated second and the three-lift concept rates third as shown below:

Pump Concept	Concept Effectiveness
Single-lift	97.9%
Two-lift	95.0%
Three-lift	90.2%

The comparative value of pump reliability of the concepts depends on the frequency and duration of outages to be expected in each. These factors, in turn, are functions of the physical configuration and operational characteristics of the concepts. Planned outages were not considered in deriving comparative reliability since the effect of planned outages are equal for all concepts.

It was decided early in the Tehachapi program planning studies that the individual pumps would be the dominating variant in comparing concept reliability and, therefore, the major investigative efforts were pointed toward an understanding of this aspect of the lift system. An absolute value of reliability must be determined after studies of valves, motors, controls, penstocks, switchgear, electrical supply and other pumping station elements yield comprehensive outage predictions for Tehachapi.

Within the scope of reliability studies of the pump, thorough analyses of statistical field data, pumping literature and wear test results revealed the following:

a. Identical pump components will wear at a rate which is approximately proportional to a power of relative water velocity.

b. In the absence of defined cavitation, repair of worn suction impellers is generally scheduled to coincide with overhaul time.

c. In view of the relative similarity of impeller wear rates for the three prototypes and design objectives which promise adequate submergence limits, it is assumed that impeller repairs coincide with pump overhauls in each lift concept.

d. It is found that multi-stage, single-suction pumps generally suffer outages in addition to those for overhaul in order that balancing labyrinths may be replaced.

e. Because of balance labyrinth repair, it appears that the single-lift prototype pumps will experience the highest frequency of scheduled outages.

f. Surveys reveal that average overhaul time for a multi-stage pump is greater than that for a single-stage pump because of a greater number of stages which must be dismantled and a greater number of components which must be maintained.

g. Wear rings with identical leakage characteristics subjected to single-stage operational parameters in the Tehachapi three-lift concept should wear at a faster rate than those in the other pump concepts because of higher head.

h. Time between pump overhauls is dictated by wear in the wearing rings so that overhaul outages in typical one and two-lift plants should be less frequent than in a typical three-lift plant used for the same application provided that identical clearance increases are tolerated.

Overall comparative reliability of the Tehachapi concepts (concept effectiveness) is the product of their ability to deliver the required supplies of water (operational reliability) and the availability of pumps to operate for the needed annual time periods (operational readiness). Operational reliability is dependent on frequency and duration of outages since concept operation involves a series of interrupted operating times with maintenance action or standby time bridging the gaps. Operational readiness is also dependent on outage frequency and duration. Application of reliability methods to the above-mentioned pump analyses reveals the following:

i. Non-coincident unscheduled outages in each plant of a multi-lift concept result in increased outage time over a single lift.

j. Increased outage in multi-lift plants reduces availability and free time which may be used for repair.

k. Forced series operation of multi-lift plant configurations together with unscheduled outages results in increased frequency of outages and decreased availability of repair time per plant.

l. The culmination of unscheduled outage effects is that the single-lift is more reliable than the two-lift which is more reliable than the three-lift concept.

Validity of the comparative reliability as derived in this preliminary report is dependent on two important factors. The first is that pump overhauls may be based more on economic life of the rings than absolute or relative clearance increases. The second is that the Tehachapi lifts are assumed to be independent of upstream plant activity.

A cursory examination of economy-based ring life as outlined in the test of this report reveals that no significant change in reliability ratings results from such analyses.

Preliminary calculations of reliability based on a mathematically correct treatment of scheduled outage frequency and repair "switching" probability indicates that no change in concept reliability ratings ensues. Procedures used in this calculations are outlined in the test of this preliminary report, but publishing of results should follow refinement of repair time distributions with regard to three shift maintenance and re-definition of scheduled and unscheduled outages with regard to series operation.

Effects of upstream plant outages on Tehachapi operations are not a part of this study but a hypothetical simulation of such effects on on Tehachapi reliability would decrease the overall reliability index and probably diminish the relative separation between the various concept indices.

Because of the extent of work which remains to be done to improve the validity of the absolute reliability indices, extreme care must be exercised in application of these factors. Numerical indices as they appear thourhgout this report must be viewed with the knowledge that their absolute accuracy is subject to qualifications but they are valid indications for comparative purposes.

In summary, it should be sufficient to point out that no reasonable variation of the parameters on which reliability is based alters the fact that the single-lift rates higher than the two-lift which rates higher than the three-lift concept.

2. Efficiency (Preliminary Test Results)

The model efficiencies recently obtained from each of the three model firms are given in Table 9-II. It must be emphasized that these are all preliminary results. The testing was performed at speeds lower than required by model test contracts for final tests and in the case of Byron Jackson, the testing was done on a commercial set-up and not on the required final set-up. Also, the Sulzer value has been corrected for mechanical losses and the other two have not. Thus, Voith and B-J values may be one or two-tenths of a percent better when this correction is applied to their results.

The Sulzer model does not have as good a surface finish as the other models have. Sulzer claims that this is consistent with their normal practice and, therefore, validates the use of their step-up formula. In Table 9-II, the stepped-up prototype efficiencies are given using the individual model test firm formulas and using the DMJM formula. The DMJM formula is thoroughly explained in Chapter 8, Volume II:

TABLE 9-II

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

Preliminary tests were made at lower rotating speeds than will be employed for final model performance tests. It was thought that the DMJM formula could be used to give comparative results. However, in view of the greater surface finish roughness on the Sulzer model, it seems best to employ the model test firm formulas for comparative purposes. The DMJM formula actually gives higher values of "stepped-up" efficiency for Voith and Byron Jackson and is very close to the Sulzer value.

Comparing the pumps and their respective lifts, the results are in the same order as originally predicted, with the single-stage pump (three-lift) being best. Comparing the model results with the predictions given in Tabel 5-II, it can be seen that the single-lift and three-lift pump types are close to the original prediction and the two-lift pump type has produced a model efficiency approximately one percent higher than predicted.

Final test results from the model program will be very valuable in substantiating the best rating on efficiency.

3. Pump, Motor, and Valve Cost

The cost of the three-lift pumps themselves is less than one-half of the single-lift pumps and less than a third of the two-lift pumps. However, because of the greater number of motors, the total motor cost for three-lifts is higher. Similarly, total valve cost is greater for the multi-lift concepts. Note: Motor and valve manufacturers have not been selected and a range of prices for various possible vendors is presented for these items. The total costs of pumps, motors and valves reflects these ranges. The total cost for the three-lift arrangement has a slight advantage over the one-lift cost and the two-lift concept is the most costly. The rating is:

Rating	Lift Concept	Cost
1st - -	Three-lift	20.185 to 22.821 million
2nd -	One-lift	22.950 to 27.870 million
3rd -	Two-lift	27.490 to 30.212 million

4. Required Submergence

A significant cost in the installation of the plants will be the excavation to supply the necessary submergence to prevent pump cavitation. The rating is:

Rating	Lift Concept	Submergence
1st -	Two-lift	approx. 64 feet) based
2nd -	One-lift	approx. 71 feet) on
3rd -	Three lift	approx. 90 feet) S = 7000

The two-lift concept is superior in this consideration and the three-lift is inferior requiring approx. 26 feet more submergence than the two-lift and 19 feet more than the one-lift when calculated from a value of S = 7000 for all concepts. (The effect of this extra excavation occurs at the first plant -- with the two-lift and three-lift systems, the upper plant ground levels can be chosen to minimize excavation work).

5. Design Details

The valves and motors are assumed to have equal design competence for each concept. With regard to the pumps themselves, each type is quite conventional in both mechanical and hydraulic design considering the special requirements of the individual pump type. Going further, a review of the Pump Detail portion of Table 9-I will show that the three pump designs are somewhat similar in their internal hydraulic design details, although there is a difference in design philosophy regarding diffuser and return channels in the multi-stage pumps. It is felt that these differences should be evaluated by comparative design testing as explained in the recommendation of Chapter 2. Excepting the comparative design testing, no rating of pumps can be made with regard to hydraulic design detail. The measured efficiency must be used for comparative evaluation of design achievement.

[illegible]